Becker

The strength and stiffness of steel under bi-axial loading

THE STRENGTH AND STIFFNESS OF STEEL UNDER BI-AXIAL LOADING

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THE STRENGTH AND STIFFNESS OF STEEL

UNDER BI-AXIAL LOADING

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PART I

INTRODUCTION

THE STRENGTH AND STIFFNESS OF STEEL UNDER BI-AXIAL LOADING

Part I

Introductory

1. General Statement.

When a steel bar is tested in tension or compression, certain phenomena are observed which are incorporated as fundamental facts in the theories of the elastic behaviour of bodies under stress. In such a test, both the strength and the stiffness of the material are observed, the former by noting the yield point and ultimate strength, the latter by observing the unit deformation corresponding to each load applied and then computing the modulus of elasticity. Repeated experiments have shown that for material of the same composition and treatment, these results are practically constant and can be used as a basis of design. The strength of any material of construction cannot be determined by mathematical analysis, neither can its stiffness. Poisson's ratio, modulus of elasticity, yield point, and Hooke's law are experimental results.

To show that strength and stiffness are entirely distinct, it is only necessary to note that the modulus of elasticity of practically all steels ranges close to thirty million pounds per square inch, while the yield point stresses and ultimate strengths vary between wide limits. Furthermore, a simple beam of span L carrying a central concentrated load W is just as strong

as the same beam with half the span carrying a central load of 2W. But the stiffness does not follow the same law, for the deflections are in the ratio of 1 to 4. The use of high carbon or vanadium steel for the arbor of a milling machine has but little advantage over an arbor of low carbon steel; the springing is a function of the modulus of elasticity which is practically constant for all grades of steel, so that two arbors of equal diameter will deflect equally under a given load, irrespective of material. But the lower yield-point steel would acquire a permanent set or deflection sooner than the other.

When, instead of dealing with a simple stress, an investigation of combined stress is attempted, the question arises whether the behaviour of the material can be accurately predicted when the calculations are based upon the values obtained in the experiments in simple tension, compression, and shear. The manner in which the elastic constants have been determined must be kept clearly in mind. Constants determined by uni-directional loading cannot be indiscriminately applied to bi-directional loading. Theories have been worked out in which these constants are used by taking account of the interaction of the applied stresses, theories, which from a mathematical standpoint are correct, but the truth or falsity of whose basic assumptions can be demonstrated only by experiment.

In developing a theory of flexure, certain assumptions are made and statements giving the ratio of applied load to both the stress and the deflection are deduced. While the stresses in a beam are the resultants of tensile or compressive stresses with shearing stresses, the fact that the shearing stresses are maximum

at the neutral axis and zero at the outer fibre, while the reverse is true of the tensile or compressive stresses, results in the stresses being but slightly, if any, in excess of the maximum tensile or compressive stresses. Experiment shows that the common theory of flexure is sufficiently close to be safely used as a basis of design. But this is true only under conditions approaching those of simple stress. Suppose the beam is circular in cross-section, carries a fly wheel near its mid-point, and at the same time transmits power. Here the effect of the combined stresses cannot be neglected and the problem of the proper method to combine the stresses arises for settlement. Can the strength and stiffness be predicted with sufficient accuracy to warrant the use of such an analysis as a basis of design? Different combinations of simple stress are possible and it may be that the same solution is not possible in all cases. The presence of shear in a bar in simple tension, and the tension and compression accompanying a torsional load shows that the governing conditions depend upon the relative strength of the material in shear, tension, and compression. A cast iron bar tested in torsion fails on an oblique plane in tension, because the tensile strength is less than the shearing strength. It is therefore logical to suppose that different stress combinations will produce failures differing in character for different materials.

2. Combined Stresses.

Three types of stress application are possible, uni-directional, bi-directional or bi-axial, and tri-directional, or stress in three directions. The first is illustrated by a specimen subjected to tension or compression in an ordinary testing

machine. Bi-directional or bi-axial stress is the application of two stresses in the same plane acting in directions at right angles to each other. Three directional stress is the application of three stresses at right angles to each other. The three types may be compared to the rectangular coordinate system with the stresses acting along one, two, or three of the coordinate axes. The condition of biaxial stress is more important, from the point of view of the engineering applications, than that of the three stresses at right angles to each other.

The possible combinations of bi-directional stress are as follows:

Tension with tension
Tension with compression
Compression with compression
Compression with tension
Shear (torsion) with tension
Shear (torsion) with compression

These may be divided into three classes, tension with tension and compression with compression forming the first, tension with compression and compression with tension the second, and the combination of either tension or compression with torsion forming the third. The third class gives but two special cases of the other two classes; for a simple torque is equivalent to two equal principal stresses, one compression and the other tension, so that a torque combined with a tension or a compression can be reduced to the case of tension combined with compression, or viceversa. To make this clearer two illustrations will be given.

A bar subjected to simple tension or compression develops

shear on oblique planes of an intensity equal to p $\sin\alpha \cos\alpha$, where p is the intensity of stress on a cross section perpendicular to the axis of the bar.

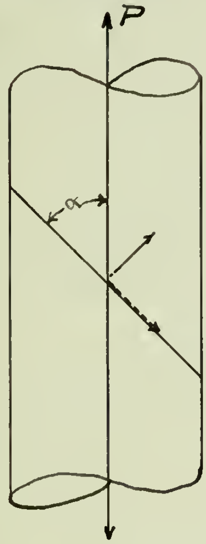

Figure 1.

The normal stress on an oblique plane is p $\sin^2\alpha$, and when $\alpha = 45°$, the shearing stress is maximum and equal to the normal stress, each being equal to $\frac{p}{2}$. An external axial tensile stress P produces a shearing stress whose maximum value is $\frac{1}{2}$p, and an axial stress whose value is $p = \frac{P}{\text{area}}$, but these do not occur in the same plane. When a bar of circular cross section is subjected to torsion the stress on a plane at right angles to the shaft is a pure shearing stress, depending in intensity upon the diameter of the bar and upon the torque. But this is not the only plane of stress. As in the bar in simple tension, there are planes on which there are both tensile and shearing stresses; there are also planes upon which there is no shearing stress.

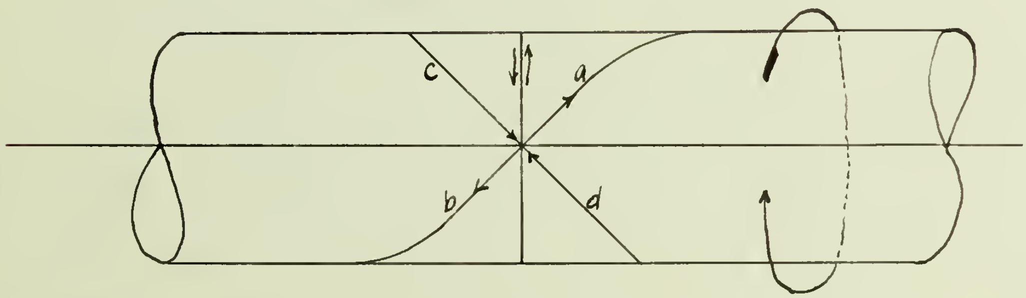

Figure 2.

Referring to Figure 2, the stresses on one 45° plane, cd, will be found to be tension, and on a plane at right angles to this one (ab, the other 45° plane), the stress is compression. With

the torque as shown in the figure, these stresses are as represented. This case is then equivalent to a bi-axial loading, with a tensile and a compressive stress, each equal to the shearing stress. This stress relation will be derived later. See page 21 .
Thus again the shearing stress is maximum on a plane making an angle of 45° with either ab or cd.

The point it is desired to bring out here is that there are stresses on oblique planes which may control the strength of the material. A good illustration of the action of these stresses is to be found in a thin tube with a slit cut along a 45° spiral. Leaving the ends uncut to apply the torque, it will be seen that if the tube is twisted in one direction the slit closes, and if it is twisted in the other direction the slit opens, but there is no tendency of the particles along the slit to move past each other.

Applications of combined stress are to be found in the familiar examples of the steam boiler for tension combined with tension, and of the crank shaft for tension or compression combined with torque. Bi-axial stresses occur in flat plates and in flat concrete slabs or girderless floors.

3. Scope of Investigation.

The purpose of this investigation is to determine the law or laws governing both the strength and stiffness of mild steel when subjected to combined stress produced by two tensions at right angles to each other. The ratio between the two tensile stresses was kept constant throughout the test of a given specimen and the comparison is made on the basis of the yield point or "apparent elastic limit" as determined by J. B. Johnson's tangent method.

The specimens tested were tubes of uniform size and

practically uniform thickness. These tubes were subjected to axial pull and internal pressure. The only variable was the ratio of axial and circumferential tensions, and comparison is made only with tubes cut from a single length of seamless drawn tubing. By means of strain gage readings a knowledge of the distribution of stress on the cross section is obtained; no assumptions are made except that of uniform distribution of the hoop tensions throughout the thickness of the tube wall.

The investigation of strength and of stiffness was carried on simultaneously, but the results are discussed separately. The points investigated are:

1. The ratio of change of yield-point stress of the material with increasing ratios of hoop tension to axial tension.
2. Stiffness of the material, strains accompanying stress, for increasing ratios of hoop tension to axial tension.

No extended discussion has been given of the engineering applications, for it is realized that while these applications are important, more work along the same lines and with other stress combinations is needed to properly establish the conclusions reached here. When this has been done and all the work has been correlated, it will be a simple matter to make an application of these principles to engineering design.

4. Acknowledgment.

All the tests were made in the Laboratory of Applied Mechanics of the University of Illinois, under the supervision of Professors A. N. Talbot and H. F. Moore, to whom acknowledgment is

made for their suggestions and criticisms, and for the interest they have shown in the progress of the investigation. Acknowledgment is also made to Mr. J. O. Draffin, Research Fellow in the Engineering Experiment Station, for his assistance in the conduct of the various tests. It is also desired to make an acknowledgment to the Joint Committee on Stresses in Railway Track for the use of the new model four-inch Berry strain gage.

PART II

THEORY

Part II

Mathematical Treatment

1. Stresses and Strains.

The mathematical theory of elasticity embodies the most complete and elaborate theory of the action of elastic bodies under stress. It will be desirable to give a brief statement of the work leading up to the derivation of the general equations of the mathematical theory of elasticity connecting stress and strain before working out the equations of stress and strain in a cylinder under axial pull and internal pressure.

The notation used will be that adopted by Love in his work on the Mathematical Theory of Elasticity.

$$\left.\begin{matrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \end{matrix}\right) \text{ strains along the } \left(\begin{matrix} \text{x axis} \\ \text{y axis} \\ \text{z axis} \end{matrix}\right.$$

$$\left.\begin{matrix} \varepsilon_{xy} \\ \varepsilon_{yz} \\ \varepsilon_{zx} \end{matrix}\right) \text{ shearing strains in a plane perpendicular to the } \left(\begin{matrix} \text{x axis} \\ \text{y axis} \\ \text{z axis} \end{matrix}\right.$$

$$\text{and parallel to the } \left(\begin{matrix} \text{y axis} \\ \text{z axis} \\ \text{x axis} \end{matrix}\right.$$

$$\left.\begin{matrix} X_x \\ Y_y \\ Z_z \end{matrix}\right) \text{ Stresses on planes perpendicular to the } \left(\begin{matrix} \text{x axis} \\ \text{y axis} \\ \text{z axis} \end{matrix}\right.$$

X_y) Shearing stresses on planes perpend- (y axis
Y_z) icular to the (z axis
Z_x) (x axis

and parallel to the (x axis
(y axis
(z axis

σ = Poissons' ratio

E = Modulus of Elasticity

Both σ and E are to be determined from tests in simple tension.

After the derivation of certain relations existing between the various strain components and similar relations between the stress components, it may be shown that, in general, the stress condition at any point of a body, loaded in any manner, is completely given by the stress ellipsoid obtained by passing through the point all possible planes and on each plane placing a vector representing the corresponding stress, both in magnitude and direction. The ends of these stress vectors lie in this stress ellipsoid. Each of these stresses makes, in general, an angle with the plane and may be resolved into normal and tangential (or shearing) components. Only in three directions at right angles to each other will there be only normal stress. These are represented by the three semiaxes of the ellipsoid and are the principal stresses for this point and condition of load.

This stress ellipsoid may be written

$$\frac{X^2}{P_1^2} + \frac{Y^2}{P_2^2} + \frac{Z^2}{P_3^2} = 1.$$

P_1, P_2, and P_3 are the principal stresses in the x, y, and z

[illegible]

directions, respectively. Here the radius is a measure of the stress on a plane perpendicular to it.

Or instead of the stress ellipsoid, the stress director quadric

$$\frac{X^2}{P_1} + \frac{Y^2}{P_2} + \frac{Z^2}{P_3} = 1.$$

may be used to determine the stress. This may be an ellipsoid, hyperboloid, etc., according as the signs of P_1, P_2, and P_3 vary. Here the radius vector from the center to any point on the quadric has the direction of the stress on a plane parallel to the tangent plane at the point.

By means of the three principal stresses--and for bi-axial loading one of these is always zero--the stresses throughout the point are completely determined. If, therefore, the principal stresses are known or can be determined from the known conditions, it is possible to determine the elastic condition of the body.

Stresses and strains are connected by means of a consideration of the work done in deforming the body, neglecting any temperature change. This gives a relation involving a force function such that its derivatives with respect to the various strains will give the stresses. At this point recourse is had to experiment to determine the nature of this force function. Hooke's law suggests a linear relation between stress and strain--within certain limits. Adopting this linear relation as general, the force function must be of the second degree and when arranged as a determinant of the quadric $\sum_1^6 c_{ij}\,\varepsilon_{ij}$ must form a symmetric determinant. For a second degree equation in six variables, the

six strains, when completely expanded will have six square terms and fifteen product terms, twenty-one in all, and consequently it must have twenty-one coefficients.

If the body is isotropic, these twenty-one coefficients can be reduced to two. Denoting by λ one of these remaining coefficients and by μ one-half the difference between the two coefficients, the following relations between stresses and strains have been established:-

$$(1).\quad X_x = \lambda\Delta + 2\mu\varepsilon_{xx}$$

$$(2).\quad Y_y = \lambda\Delta + 2\mu\varepsilon_{yy}$$

$$(3).\quad Z_z = \lambda\Delta + 2\mu\varepsilon_{zz}$$

where $\Delta =$ the dilatation $= \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz}$

$$(4).\quad Z_y = Y_z = \mu\varepsilon_{yz}$$

$$(5).\quad X_z = Z_x = \mu\varepsilon_{zx}$$

$$(6).\quad Y_x = X_y = \mu\varepsilon_{xy}$$

Applying these relations to a bar in simple tension, the relation between Poisson's ratio and the modulus of elasticity is established.

$$(7)\quad E = \mu\frac{3\lambda + 2\mu}{\lambda+\mu}$$

$$(8)\quad \sigma = \frac{\lambda}{2(\lambda+\mu)}$$

$(9)\quad G =$ *modulus of shearing elasticity* $= \mu$.

From these the relation between E and G may be derived.

$$G = \frac{E}{2(1+\sigma)}$$

or, if $m = \frac{1}{\sigma}$,

$$G = \frac{1}{2}\left(\frac{m}{m+1}\right)E.$$

The values of λ and μ may now be established in terms of E and σ.

$$(10) \qquad \lambda = \frac{E\sigma}{(1+\sigma)(1-2\sigma)}$$

$$(11) \qquad \mu = \frac{E}{2(1+\sigma)}$$

Adding equations (2) and (3)

$$Y_y + Z_z = 2\lambda\Delta + 2\mu(\varepsilon_{yy} + \varepsilon_{zz})$$
$$= 2\lambda\Delta + 2\mu(\Delta - \varepsilon_{xx})$$

$$\Delta = \frac{Y_y + Z_z + 2\mu\varepsilon_{xx}}{2(\lambda+\mu)}$$

$$\lambda\Delta = \frac{\lambda(Y_y + Z_z + 2\mu\varepsilon_{xx})}{2(\lambda+\mu)}$$

$$= \sigma(Y_y + Z_z + 2\mu\varepsilon_{xx})$$

$$X_x - 2\mu\varepsilon_{xx} = \sigma(Y_y + Z_z) + \mu\varepsilon_{xx}\frac{\lambda}{\lambda+\mu}$$

$$X_x = \sigma(Y_y + Z_z) + \varepsilon_{xx}\left(\frac{3\lambda+2\mu}{\lambda+\mu}\right)$$

$$= \sigma(Y_y + Z_z) + E\varepsilon_{xx}$$

Similarly,

$$Y_y = \sigma(Z_z + X_x) + E\varepsilon_{yy}$$
$$Z_z = \sigma(X_x + Y_y) + E\varepsilon_{zz}$$

Rearrangement of the above gives,

$$E\varepsilon_{xx} = X_x - \sigma(Y_y + Z_z)$$
$$E\varepsilon_{yy} = Y_y - \sigma(X_x + Z_z)$$
$$E\varepsilon_{zz} = Z_z - \sigma(X_x + Y_y).$$

These are the three fundamental equations connecting stress and strain. $E\varepsilon_{xx}$, $E\varepsilon_{yy}$, and $E\varepsilon_{zz}$ are called by various writers the reduced stresses, the true stresses, or the ideal stresses, but in this discussion the term 'reduced stresses' will be used. These

three equations form the basis of the maximum strain theory.

2. Stresses and Strains in a Thin Tube.

Changing the coordinate system from rectangular to cylindrical coordinates, the following are the expressions for the strains:-

$$\varepsilon_{rr} = \frac{\partial \overline{U}_r}{\partial r}$$

$$\varepsilon_{\theta\theta} = \frac{1}{r}\frac{\partial \overline{U}_\theta}{\partial \theta} + \frac{\overline{U}_r}{r}$$

$$\varepsilon_{\varphi\varphi} = \frac{1}{r \sin\varphi}\frac{\partial \overline{U}_\varphi}{\partial \varphi} + \frac{\overline{U}}{r}$$

and for the stresses

$$\widehat{rr} = \lambda\Delta + 2\mu\,\varepsilon_{rr}$$

$$\widehat{\theta\theta} = \lambda\Delta + 2\mu\,\varepsilon_{\theta\theta}$$

$$\widehat{\varphi\varphi} = \lambda\Delta + 2\mu\,\varepsilon_{\varphi\varphi}$$

where ε_{rr} is the strain in a radial direction,

$\varepsilon_{\theta\theta}$ is the strain in a circumferential direction,

$\varepsilon_{\varphi\varphi}$ is the strain in the direction of the third coordinate.

For a cylinder $\varepsilon_{\varphi\varphi}$ becomes ε_{zz}.

$\widehat{rr}$ is the stress in a radial direction,

$\widehat{\theta\theta}$ is the stress in a circumferential direction,

$\widehat{\varphi\varphi}$ is the stress in the direction of the third coordinate.

$\widehat{\varphi\varphi}$ becomes $\widehat{zz}$ for a cylinder.

$\overline{U}$ is the displacement of the point, and its components are $\overline{U}_r$, $\overline{U}_\theta$, and $\overline{U}_\varphi$.

For a cylinder under internal pressure only, with no externally applied axial tension or compression, the strain components reduce to the following, since $\overline{U}_\theta$ and $\overline{U}_\varphi$ are equal to zero and hence $\overline{U} = \overline{U}_r$.

$$\varepsilon_{rr} = \frac{\partial \overline{U}}{\partial r}$$

$$\varepsilon_{\theta\theta} = \frac{U}{r}$$

$$\varepsilon_{zz} = \frac{\partial w}{\partial z}$$

$$(\text{Since } U_\theta = 0), \text{ and}$$

$$\Delta = \frac{\partial U}{\partial r} + \frac{U}{r} + \frac{\partial w}{\partial z}$$

Now assuming that plane cross sections remain plane, the displacements along the z axis will be proportional to z; that is,

$$w = az$$

$$\frac{\partial w}{\partial z} = a$$

$$\frac{\partial w}{\partial r} = 0 .$$

The stress components are

$$\widehat{rr} = \lambda\Delta + 2\mu\varepsilon_{rr}$$

$$\widehat{\theta\theta} = \lambda\Delta + 2\mu\varepsilon_{\theta\theta}$$

$$\widehat{zz} = \lambda\Delta + 2\mu\varepsilon_{zz} .$$

The equilibrium equation is

$$(\lambda + 2\mu)\frac{\partial\Delta}{\partial r} + 2\mu\frac{\partial\omega}{\partial z} + \rho F_r = 0,$$

or neglecting the body forces (weight of the tube) and substituting for Δ, since the rotation ω is zero,

$$(\lambda + 2\mu)\frac{\partial}{\partial r}\left(\frac{\partial U}{\partial r} + \frac{U}{r} + a\right) = 0.$$

Integrating twice,

$$U_r = \tfrac{1}{2}Cr^2 + D$$

$$U = Ar + \frac{B}{r}$$

$$\therefore \widehat{rr} = \lambda\Delta + 2\mu\varepsilon_{rr}$$

$$= (\lambda + 2\mu)\left(A - \frac{B}{r^2}\right) + \lambda\left(A + \frac{B}{r^2} + a\right).$$

If the internal pressure of the cylinder is p_1, the internal raduis is r_1, the external pressure is zero, and the external radius is r_0, then this equation gives at the inner and outer surfaces,

$$-p_1 = 2A(\lambda+\mu) - 2\mu\frac{B}{r_1^2} + \lambda a$$

$$0 = 2A(\lambda+\mu) - 2\mu\frac{B}{r_0^2} + \lambda a$$

from which

$$2\mu B = p_1\left(\frac{r_1^2 r_0^2}{r_0^2 - r_1^2}\right) = T$$

and

$$2A(\lambda+\mu) = \frac{p_1 r_1^2}{r_0^2 - r_1^2} - \lambda a = S - \lambda a.$$

Let $\widehat{zz} = KS$; that is, an axial stress is added proportional to the hoop tension, such that K is equal to 1 when the axial tension is one half the hoop tension. K, therefore, is equal to twice the reciprocal of the ratio of the hoop tension to the axial tension.

Then

$$KS = \lambda\Delta + 2\mu\varepsilon_{zz} = \lambda\Delta + 2\mu a$$

$$= \lambda\left(\frac{\partial U}{\partial r} + \frac{U}{r} + a\right) + 2\mu a$$

$$= \lambda(2A + a) + 2\mu a$$

$$= \frac{S\lambda + (3\lambda + 2\mu)\mu a}{\lambda + \mu}$$

from which

$$a = \frac{S(K(\mu+\lambda) - \lambda)}{\mu(3\lambda + 2\mu)}.$$

$$\varepsilon_{\theta\theta} = \frac{U}{r} = A + \frac{B}{r^2} = \frac{S - \lambda a}{2(\lambda+\mu)} + \frac{T}{2\mu r^2}$$

$$= \frac{S}{2(\lambda+\mu)} - \frac{\lambda S(K(\mu+\lambda) - \lambda)}{2\mu(3\lambda+2\mu)(\lambda+\mu)} + \frac{T}{2\mu r^2}$$

$$= \frac{1}{E}\left[\frac{S((3\lambda+2\mu)\mu - \lambda(K(\lambda+\mu) - \lambda))}{2(\lambda+\mu)^2}\right] + \frac{T}{2\mu r^2}$$

$$= \frac{1}{E}\left[S((1-2\sigma)(1+\sigma) - \sigma(K - 2\sigma)) + \frac{T}{r^2}(1+\sigma)\right]$$

$$= \frac{1}{E}\left[S(1 - \sigma(1+K)) + \frac{T}{r^2}(1+\sigma)\right]$$

$$\varepsilon_{\theta\theta} = \frac{1}{E}\left[\frac{p_1 r_1^2}{r_0^2 - r_1^2}\left[1-\sigma(1+K)\right] + \frac{p_1 r_0^2 r_1^2}{r^2(r_0^2 - r_1^2)}(1+\sigma)\right]$$

Setting $r = r_0$, the stresses and strains obtained are those on the outside of the cylinder. When $r = r_1$, the stresses and strains are those on the inside of the cylinder. Since all measurements of strains were made on the outside of the tube, the values of $r = r_0$ will be used.

With $r = r_0$, the above equation reduces to

$$\varepsilon_{\theta\theta} = \frac{1}{E}\frac{p_1 r_1^2}{r_0^2 - r_1^2}\left[2 - \sigma K\right]$$

$$\varepsilon_{zz} = \frac{\partial w}{\partial z} = a = \frac{S(K(\lambda+\mu)-\lambda)}{\mu(3\lambda+2\mu)} = \frac{S}{E}(K-2\sigma)$$

$$= \frac{1}{E}\frac{p_1 r_1^2}{r_0^2 - r_1^2}(K-2\sigma)$$

which is the axial elongation.

Now $\pi p_1 r_1^2 = P$ is the axial load on the tube due to the hydrostatic pressure and $\pi(r_0^2 - r_1^2) = A$ is the area of the tube, hence

$$\varepsilon_{\theta\theta} = \frac{1}{E}\frac{P}{A}\left[2-\sigma K\right]$$

$$\varepsilon_{zz} = \frac{1}{E}\frac{P}{A}\left[K - 2\sigma\right]$$

But $K P = L$, where L is the total axial load.

$$\therefore \quad \varepsilon_{\theta\theta} = \frac{L}{EA}\left[\frac{2-\sigma K}{K}\right]$$

$$\varepsilon_{zz} = \frac{L}{EA}\left[\frac{K-2\sigma}{K}\right]$$

The corresponding reduced stresses are

$$E\varepsilon_{\theta\theta} = \frac{L}{A}\left[\frac{2}{K} - \sigma\right]$$

$$E\varepsilon_{zz} = \frac{L}{A}\left[1 - \frac{2\sigma}{K}\right] \qquad \left(\frac{L}{A} = \text{the unit axial stress}\right).$$

and the apparent or applied stresses are

$$\widehat{zz} = KS$$

$$= \frac{K p_1 r_1^2}{r_0^2 - r_1^2} = \frac{L}{A}.$$

$$\widehat{\theta\theta} = \lambda\Delta + 2\mu\varepsilon_{\theta\theta}$$
$$= \lambda\left(\frac{\partial U}{\partial r} + \frac{U}{r} + a\right) + \frac{2\mu U}{r}$$
$$= (\lambda + 2\mu)\frac{U}{r} + \lambda\frac{\partial U}{\partial r} + \lambda a$$
$$= 2\mathcal{A}(\lambda+\mu) + \frac{2\mu B}{r^2} + \lambda a$$
$$= S + \frac{T}{r^2}$$
$$= \frac{2p_1 r_1^2}{r_0^2 - r_1^2}$$
$$= \frac{2P}{A}$$
$$= \frac{2L}{KA}.$$

$$\widehat{rr} = \lambda\Delta + 2\mu\varepsilon_{rr}$$
$$= \lambda\left(\frac{\partial U}{\partial r} + \frac{U}{r} + a\right) + 2\mu\frac{\partial U}{\partial r}$$
$$= (\lambda + 2\mu)\frac{\partial U}{\partial r} + \frac{\lambda U}{r} + \lambda a$$
$$= 2\mathcal{A}(\lambda+\mu) - \frac{2\mu B}{r^2} + \lambda a$$
$$= S - \frac{T}{r^2}$$
$$= 0$$

Investigating the conditions on the inside of the tube, the axial strains ε_{zz} are seen to be the same; while, since the ratio of r_1 to r_0 is .97, the circumferential strains are increased about 3%. The stresses are affected to the same extent, so that the radial stress $\widehat{rr}$ is given by

$$\frac{p_1 r_1^2}{r_0^2 - r_1^2} - \frac{p_1 r_0^2}{r_0^2 - r_1^2} = p_1, \qquad \text{the internal}$$

hydrostatic pressure.

3. Poisson's ratio.

In a cylindrical bar subjected to combined axial load and torsion, there is a normal stress t and a shearing stress s on a plane perpendicular to the axis of the bar, and an equal shearing stress s on a plane parallel to the axis of the bar. The principal stresses in this case are*

$$t_1 = \frac{t}{2} + \sqrt{\frac{t^2}{4} + s^2}$$

$$t_2 = \frac{t}{2} - \sqrt{\frac{t^2}{4} + s^2}$$

Let ε_1 and ε_2 denote the strains in the directions of the principal stresses.

Then

$$E\varepsilon_1 = t_1 - \sigma t_2$$

$$E\varepsilon_2 = t_2 - \sigma t_1$$

from which, by substituting for t_1 and t_2 their values,

$$E\varepsilon_1 = \frac{1-\sigma}{2}t + \frac{1+\sigma}{2}\sqrt{t^2 + 4s^2}$$

$$E\varepsilon_2 = \frac{1-\sigma}{2}t - \frac{1+\sigma}{2}\sqrt{t^2 + 4s^2}$$

* Lanza, Applied Mechanics, 1910, page 878

These are the greatest and least reduced stresses.

Solving the equation of the greatest stress for s,

$$s^2 = \frac{(E\varepsilon_1)^2 - \sigma t^2 - E\varepsilon_1(1-\sigma)t}{(1+\sigma)^2},$$

while, when t = o

$$s = \frac{E\varepsilon_1}{1+\sigma}$$

and when s = o

$$t = E\varepsilon_1.$$

Therefore, in order that these equations shall hold for all possible values of s and t, the ratio of the shearing yield point stress to the tension yield point stress should be

$$\frac{s}{t} = \frac{\frac{E\varepsilon_1}{1+\sigma}}{E\varepsilon_1} = \frac{1}{1+\sigma},$$

when

$$\sigma = \frac{1}{4}, \qquad \frac{s}{t} = 0.80$$

$$\sigma = \frac{1}{3}, \qquad \frac{s}{t} = 0.75$$

$$\sigma = \frac{1}{2}, \qquad \frac{s}{t} = 0.66.$$

It remains for experiment to show whether these deductions are true.

σ cannot be greater than one half, for if it could be, a body would contract under three equal tensions. See equation (a) page 34.

4. Shear.

It can be shown* that the shear s on any plane whose normal makes an angle α with the X axis is, for the case of two principal stresses t_1 and t_2,

* Lanza, Applied Mechanics, 1910 page 880

$$s = (t_1 - t_2)\cos\alpha \sin\alpha,$$

which is maximum for $\alpha = 45°$ and

$$s_{max} = \frac{1}{2}(t_1 - t_2)$$

If now t_1 and t_2 are of opposite signs, the shearing stress is equal to one half their sum, while if one principal stress is zero the shearing stress is equal to one half the other principal stress. As in the tubes the third principal stress is zero or very small, the maximum shearing stresses occur on planes containing one of the other principal stresses and cutting the remaining one at an angle of 45°. When the stress ratio is 1 to 1, these maximum shearing stresses will be equal, otherwise the maximum shearing stress will be determined by the greatest principal stress, will be equal to one half this stress, and act on a plane containing the second principal stress and cutting the first one at an angle of 45°.

5. Theories of Combined Stress.

Six theories have been advanced to cover the problems of combined stress. Two of them are empirical, one is developed from a molecular hypothesis, one from the mathematical theory of elasticity, and two from static relations of stresses. Three of these theories have found considerable favor and are given first. The last three are introduced, not because they are considered important, but with the view of covering the field as completely as possible.

1. Maximum Strain Theory. The maximum strain theory takes the equations for the reduced stress derived by the mathematical theory of elasticity, (see page 15) and assumes

[illegible]

that whatever the combination of stresses, the material will fail when the maximum strain (which will be in the direction of the greatest stress) reaches a value equal to that at the yield-point stress in simple tension or compression. At the yield-point stress Ee must be a constant in all cases. The maximum strain theory holds that when a material is acted upon by two or three stresses at right angles to each other, its strength is increased if the stresses are of like sign and that its strength is diminished if the stresses are opposite in sign. Thus two tensions or two compressions will produce an increase in strength, whereas a tension combined with a compression produces a reduction of strength. For a stress ratio of one to one, both stresses tension, the material will be increased in strength 43% if $\sigma =$ .3, while if one stress is tension and the other compression, it will be weakened 23% for the same stress ratio.

If in equation (a) (see page 15) Yy and Zz are zero, the case is that of a bar in simple tension, (compression is expressed as negative tension) and dividing both sides of the equation by ε_{xx}, the result is the equation of the modulus of elasticity. For combined stress, then, the strain accompanying a given stress is changed by the addition of another stress at right angles to the first. It is increased if the stresses have unlike signs and diminished if they have like signs. Also, since Ee is called the reduced stress, the strain ε is the measure of this stress and the material will not reach the yield point until this strain reaches the value corresponding to the strain obtained in simple tension at the yield point. It should be em-

phasized that all elastic theory holds only within the elastic limit, or more correctly within the limit of proportionality, where E remains constant. But the slight variation up to the yield point, even though the value of E does change slightly, does not invalidate the theory, and the yield point is commonly taken as the limit of the discussion.

The maximum strain theory is developed from the mathematical theory of elasticity. Its weakness lies in the connecting link between stresses and strains, where temperature effect is neglected and where Hooke's law is assumed to hold rigidly and is taken as the basis of the degree of the force function. It has been shown* that there is a cooling of a bar of metal as the stress is increased up to the yield point, and it is also well known that Hooke's law is only an approximation. A very good approximation it is, to be sure, for engineering purposes, but lack of isotropy in the materials, cold working, and similar causes tend to change conditions, so that a slight deviation from Hooke's law may be observed considerably before the yield point is reached. While the mathematical theory of elasticity gives a good foundation for the maximum strain theory, or the theory of St. Venant, as it is often called, it must not be expected that the measured strains upon a body known to be not wholly isotropic, will conform exactly to this theory of stiffness.

The question of strength is quite a different one, for there is no assurance that the strains are the true measures of strength. Reasonable as the assumption may be, it is an assumption,

*C. A. P. Turner, Trans. Am.Soc.C. E. 1902
Lawson and Capp, Inter.Assn. Test. Mat. 1912
Ew. Rasch, Inter. Assn. Test. Mat., 1909

whose truth or falsity must be demonstrated by experiment.

2. The Maximum Stress Theory. The maximum stress theory, or Rankine's theory as it is sometimes called, holds that whatever the ratio of the stresses in two directions, and whether they are of like or opposite signs, the material will reach the yield point when, and only when, one of the stresses reaches the value corresponding to the yield-point stress in simple tension or compression as the case may be. It takes no account of Poisson's ratio as affecting strength, and states that a material is neither weakened nor strengthened by the addition of a second stress at right angles to the first. If, then, this theory holds, the material should reach its yield point when the greater stress reaches the yield-point stress for uni-directional loading.

3. Maximum Shear Theory. In the preceding theories, failure is considered to take place in tension or compression, whereas this theory holds that all failures are failures by yielding in shear when the shearing unit-stress reaches the shearing yield-point stress. Therefore, if the maximum shearing stresses are the same for corresponding loads in simple stress and combined stress, the failure will be identical in the two cases.

The basic principle of the maximum shear theory, that the failure in combined stress is the result of the shearing stress reaching the yield point, when carried to the logical conclusion demands that when two principal stresses are zero the failure is still due to shear. A steel bar subjected to axial tension only must therefore fail in shear. It has been shown (see page 22)

that the maximum shear occurs on a 45° plane and that its intensity is one half the tensile unit stress. If the failure occurs equally in both ways --tension and shear -- the yield-point stress in shear must be just one half that in tension, and if the shearing yield-point stress is reached first--as this theory maintains--then the ratio is somewhat less than one half.

If the combined stresses are a compression and a tension, the resulting maximum shearing stress is one half the sum of the tensile and compressive unit stresses. When these tensile and compressive stresses are equal, the intensity of the shearing stress is equal to the intensity of the tensile or compressive stresses, and failure will take place by shear unless the shearing yield-point stress is equal to or greater than that of either tension or compression. It seems entirely possible, then, that failure may be caused under certain conditions by shear, and that in other cases its intensity may be insufficient to cause yielding.

Considering compression as negative tension, there are two kinds of elementary stress treated in mechanics, tension and shear. They are quite distinct and have different accompanying phenomena. While a definite relationship may be established between the shearing and tensile stresses, the material may fail either in tension or in shear. Mild steel in torsion gives a square break, a shearing failure, but cast iron tested in torson breaks along a helicoid, failing in tension because the material is weaker in tension than in shear.

This duality of conditions, while not entirely overlooked, has been advanced solely to form two distinct theories of failure which have never been connected. The possibility that both shear

and tension may govern, each within certain limits, has apparently not been mentioned.

The usual stress derivation for combined stress given in text books is based upon the static equilibrium of forces.

Let p = the unit stress due to tension or bending moment
" q = " " shear " " torque
Let p_1 = the resultant normal stress
" q_1 = " " shearing "

Then, since the problem is ordinarily applied to a circular shaft in combined bending and torsion, the acting stresses are as shown in Figure 3.

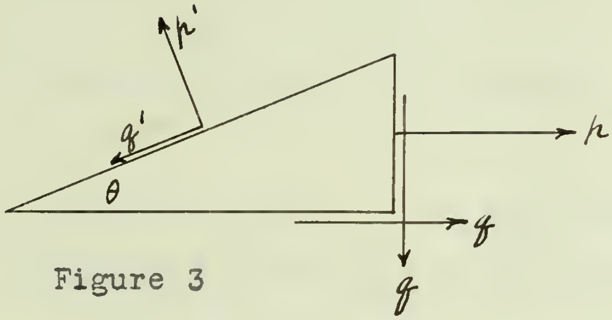

Figure 3

If the area over which p_1 and q_1 act is a, the area on which p acts is a sin θ and the horizontal shear acts on an area a cos θ.

Taking components along a line at right angles to the diagonal,

$$p'a - pa\sin^2\theta - qa\sin\theta\cos\theta - qa\sin\theta\cos\theta = 0$$

$$p' = \tfrac{1}{2}p\sin^2\theta - 2q\sin\theta\cos\theta$$

$$p' = \tfrac{1}{2}p(1-\cos 2\theta) + q\sin 2\theta$$

$$\frac{dp'}{d\theta} = p\sin 2\theta + 2q\cos 2\theta.$$

For maximum p',

$$\tan 2\theta = -\frac{2q}{p},$$

so that solving for $\sin 2\theta$ and $\cos 2\theta$,

$$p'_{max.} = \tfrac{1}{2}(p + \sqrt{p^2 + 4q^2}$$

Taking components along the diagonal and substituting

the value of θ for q_1 max,

$$q'_{max.} = \frac{1}{2}\sqrt{p^2+4q^2}.$$

Safe working stresses are assigned for p_1 and q_1 and the two formulas transformed into terms of the torque, bending moment, and radius of the shaft. The larger of the two results is to be used. By assigning values to p_1 and q_1 for safe working stresses, the ratio of shear to tension is fixed, and this constitutes the nearest approach to the recognition of a dual law.

4. Internal Friction Theory. It has been said that the chief difference between the internal friction theory and the maximum shear theory is that the former is based upon a minimum resistance to sliding, while the latter is based upon a maximum shearing stress. If the angle of friction is zero, the internal friction theory becomes the maximum stress theory.

A short cylinder of brittle material when tested in compression, fractures by shearing along a diagonal plane which should make an angle of 45° with the axis, since this is the plane of greatest shearing intensity. But the angles observed in experiments are always different from 45°. The attempt to explain this variation resulted in the theory of internal friction, first propounded by Navier. When two particles, acted upon by a stress, tend to slide over each other, a condition is set up similar to that of ordinary sliding friction. On the supposition that this resistance is similar to sliding friction, one of the

laws governing the latter is applied; namely, that the coefficient of internal friction is independent of the load or stress. Therefore slipping will occur along the surface of the plane inclined at an angle β, (Figure 4) such that

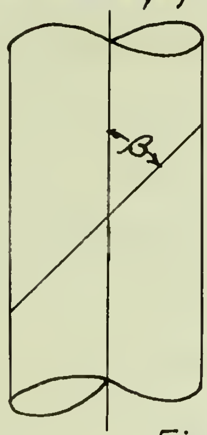

Fig. 4.

$$\beta = 45^\circ - \frac{\varphi}{2} \quad \text{for compression}$$

and

$$\beta = 45^\circ + \frac{\varphi}{2} \quad \text{for tension}$$

φ is the angle of friction and $\mu = \tan\varphi$ the coefficient of friction

Let t = the yield-point stress in tension

Let c = the yield-point stress in compression.

If the limiting friction per unit of surface is the same in each case, then the normal stress on the surface of slipping, at the instant when yielding begins, must be the same for each, since this is $\frac{1}{\mu}$ times the limiting friction.

Let the value of the normal stress be n. For tension, the surface of slipping is inclined to the axis of the bar at an angle $\beta = 45 + \frac{\varphi}{2}$ and

$$n = \frac{t}{2}\sin^2\left(45 + \frac{\varphi}{2}\right) = \frac{t}{2}(1 + \sin\varphi).$$

For compression, the surface of slipping is inclined at an angle $\beta = 45 - \frac{\varphi}{2}$ and

$$n = \frac{c}{2}(1 - \sin\varphi).$$

Hence

$$\frac{t}{c} = \frac{1 - \sin\varphi}{1 + \sin\varphi} = \frac{\sqrt{1+\mu^2} - \mu}{\sqrt{1+\mu^2} + \mu} = \frac{1-\mu}{1+\mu}. \text{ (approx.)}$$

If $\varphi = 10^\circ$, $\mu = .176$ and $\frac{t}{c} = .701$.

The ratio between the tension and compression yield points required by this theory is perhaps its weakest point as

far as ductile materials are concerned. All attempts at experimental verification have thus far failed and likewise attempts to fit combined stress experimental results to this formula have have not given satisfactory results. Gulliver* has attempted to amplify the theory by the addition of a factor depending upon cohesion of the particles, but this is an obvious attempt to bolster it up to fit the experimental data. The theory has not been accepted for ductile materials by engineers, and no special emphasis is placed upon it here.

The four theories just outlined may be combined into one general formula. If the equivalent tensile or compressive stress is t', then it may be expressed as the following function of the simple tension t and the shear s

$$t' = \frac{at}{2} + \frac{b}{2}\sqrt{t^2+4s^2}$$

By changing the constants a and b, each of the four theories may be obtained.

For use in designing, the four theories are frequently written in terms of the applied torque and bending moment. This gives for the maximum strain theory,

$$M_{(eq)} = \frac{3}{8}M + \frac{5}{8}\sqrt{M^2+T^2}$$
$$(\sigma = .25)$$

For the maximum stress theory,

$$M_{(eq)} = \frac{1}{2}M + \frac{1}{2}\sqrt{M^2+T^2}$$

For the maximum shear theory,

$$T_{(eq)} = M_{(eq)} = \sqrt{M^2+T^2}$$

For the internal friction theory,

$$M_{(eq)} = \frac{1}{1+\sin\varphi}\left[M\sin\varphi + \sqrt{M^2+T^2}\right]$$

* Proc. Royal Society of Edinburgh, 1907-08

M(eq.) is the equivalent bending moment and requires the use of a unit stress in tension or compression.

T(eq.) is the equivalent torque and requires the use of a unit stress in shear.

It is seen that the torque and the bending moment (each acting alone) required to produce failure has the same numerical value according to the maximum shear theory. This follows logically from the contention that the shearing yield-point stress is one half the tensile yield-point stress.

Writing these equations in a general form,

$$M_{(eq)} = CM + D\sqrt{M^2+T^2}$$

5. Mohrs' theory.*

Let K_4 = the shearing yield-point stress,

" K_3 = the stress in compression and tension (equal) which together produce a shearing stress equal to the shearing yield-point stress K_4.

Let K_1 = the tension yield-point stress,

" K_2 = the compressive yield-point stress.

Then

$$K_3 = \frac{K_1 K_2}{K_1 + K_2}$$

$$K_4 = \frac{1}{2}\sqrt{K_1 K_2}$$

This theorem strikes at the usual assumption that the shear in a material has a fixed relation to the tension or compression, by making it a function of both properties instead of either alone. The usual theory gives for two equal stresses of unlike sign the relation

* Zeitschrift des Vereins Deutscher Ingenieure, 1900. page 1530

Shear = $\frac{1}{2}$ (tension + compression)

or in the above notation

$$K_4 = \frac{1}{2}(K_3 + K_3) = K_3$$

Applying this to the formulas above,

$$\frac{1}{2}\sqrt{K_1K_2} = \frac{K_1K_2}{K_1+K_2}$$

from which

$$\frac{1}{4}K_1^2 + \frac{1}{4}K_2^2 + \frac{1}{2}K_1K_2 = K_1K_2,$$

$$K_1 - K_2 = 0,$$

or the usual theory is a special case of Mohr's theory which assumes the tensile and compressive yield-point stresses equal. It appears, then, that Mohr's theory is an attempt to modify the shear relation according to the physical properties of the material. If the yield-point stresses for tension and compression are the same, this theory presents nothing new, for it coincides with the maximum shear theory. If the yield-point stresses are different, it brings in a new relation regarding the shear failure in combined stress. It takes no cognizance, however, of the variation in shearing yield-point stresses in various directions throughout the material. This theory, like that of internal friction, has found little acceptance. The author's experimental verification is based almost wholly upon rupture conditions. It is virtually an acceptance of the maximum shear theory with the definition of the value of that shear at the yield point.

6. Wehage's Theory.* This is literally a paper theory, as it is based upon a few experiments in cross shaped pieces of paper submitted to tension in two directions at right angles to each other. If the material has a different yield-point stress

* Zeitschrift des Vereins Deutscher Ingenieure 1905. page 1077.

in two directions the following elliptic relation is given as an empirical deduction:

$$\left(\frac{t_1}{T_1}\right)^2 + \left(\frac{t_2}{T_2}\right)^2 = 1.$$

T_1 and T_2 are the yield-point or ultimate stresses in the two directions (as, for instance, with and across the direction of rolling), and t_1 and t_2 the applied unit stresses in the corresponding directions, when $T_1 = T_2$ the ellipse becomes a circle.

This theory assumes that the material is weakened by the application of two tensions for the reason that such stresses tend to lessen the cohesion between the fibres. The assertion is also made that a compression combined with a tension should strengthen the material by increasing this cohesion, although no formula is proposed. What kind of a formula could be made to fit this rather unique idea is not evident.

Recapitulating, the mathematical statement of the six theories is given by the following equations:

1. Maximum Strain Theory.

$$(a)\quad E\varepsilon_{xx} = X_x - \sigma(Y_y + Z_z)$$

$$(b)\quad E\varepsilon_{yy} = Y_y - \sigma(X_x + Z_z)$$

$$(c)\quad E\varepsilon_{zz} = Z_z - \sigma(X_x + Y_y)$$

$$(d)\quad t' = \frac{1-\sigma}{2}t + \frac{1+\sigma}{2}\sqrt{t^2+4s^2}$$

2. Maximum Stress Theory.

$$t' = \frac{1}{2}t + \frac{1}{2}\sqrt{t^2+4s^2}$$

3. Maximum Shear Theory.

$$s' = \frac{1}{2}\sqrt{t^2+4s^2}$$

4. Internal Friction Theory.

$$s' = \frac{t}{2}\tan\varphi + \sec\varphi\sqrt{\frac{t^2}{4}+s^2}$$

5. Mohr's Theory.

$$K_4 = \frac{1}{2}\sqrt{K_1 K_2}$$

$$K_3 = \frac{K_1 K_2}{K_1 + K_2}$$

6. Wehage's Theory.

$$\left(\frac{t_1}{T_1}\right)^2 + \left(\frac{t_2}{T_2}\right)^2 = 1,$$

or if the material is isotropic,

$$t_1^2 + t_2^2 = T^2.$$

7. Graphical Presentation of Three Theories. A graphical presentation usually serves to give a better idea of the working of a theory or formula, and for this reason the three most important theories are represented for the four combinations of simple tension or compression. To make it more general, a different yield-point stress in compression and tension has been assumed for the maximum stress theory, which is the only theory which permits it.

Maximum Strain Theory

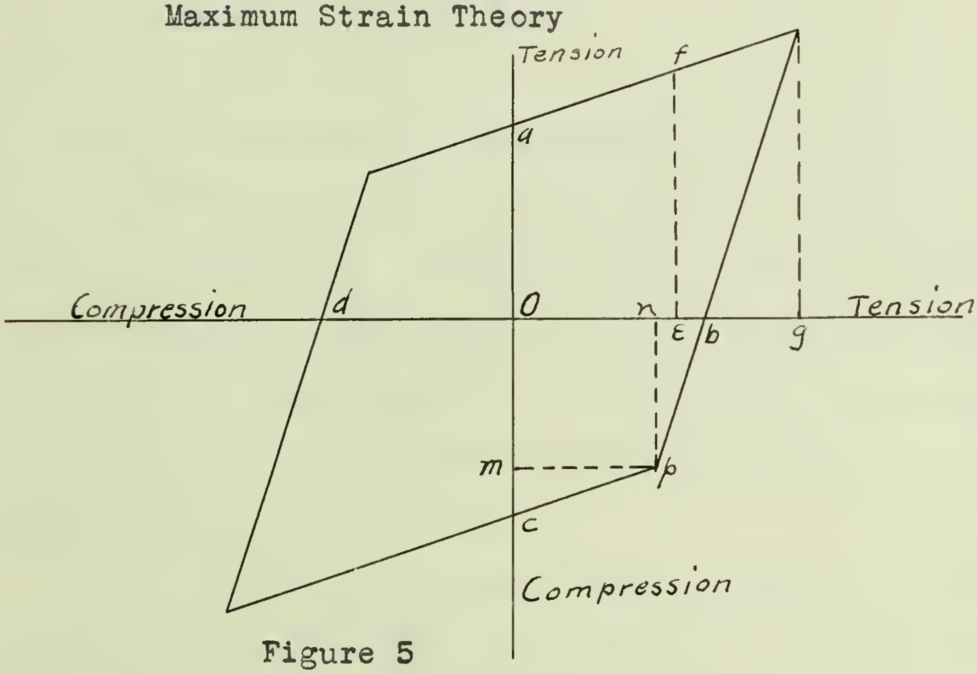

Figure 5

Let Oa (Figure 5) and Ob represent the yield-point stress in simple tension and Oc and Od that in compression. The

maximum strain theory requires that Oc = Ob; that is, that the yield-point stresses in tension and compression shall be the same. For, take two equal stresses, one compression and the other tension, sufficient to produce yielding in either tension or compression, whichever is the weaker. Then from the equations (a) and (b), page 34,

$$E\varepsilon_{xx} = X_x - \sigma Y_y$$
$$E\varepsilon_{yy} = Y_y - \sigma X_x,$$

or in the more usual notation,

$$E\varepsilon_1 = \sigma_1 - \frac{\sigma_2}{m}$$
$$E\varepsilon_2 = \sigma_2 - \frac{\sigma_1}{m}.$$

Since $\sigma_1 = \sigma_2$ by hypothesis; that is, the tensile and compressive stresses are equal, and equal to the yield-point stress in either tension or compression, whichever is the weaker,

$$E\varepsilon_1 = \sigma_1\left(1 - \frac{1}{m}\right)$$
$$E\varepsilon_2 = \sigma_1\left(1 - \frac{1}{m}\right)$$

Hence $\quad E\varepsilon_1 = E\varepsilon_2$.

But Ee_1 is the tensile yield-point stress and Ee_2 is the compression yield-point stress, hence they are the same. Again referring to Figure 5, a tensile stress equal to Oe would require a tensile stress equal to ef at right angles to cause yielding. For two equal stresses the condition of yielding would not be reached until each stress attained the value og. The increase in strength is therefore og − ob.

For a compression combined with a tension, the material would be weakened, so that for equal stresses in tension and compression, yielding would occur when each equalled on = om. The

two other quadrants are similar, two compressions producing the same relative effect as two tensions, and a tension and compression producing a corresponding effect to a compression and a tension.

Maximum Stress Theory.

Since the greater stress controls, this diagram will be a square whose center is not at the center of the coordinates.

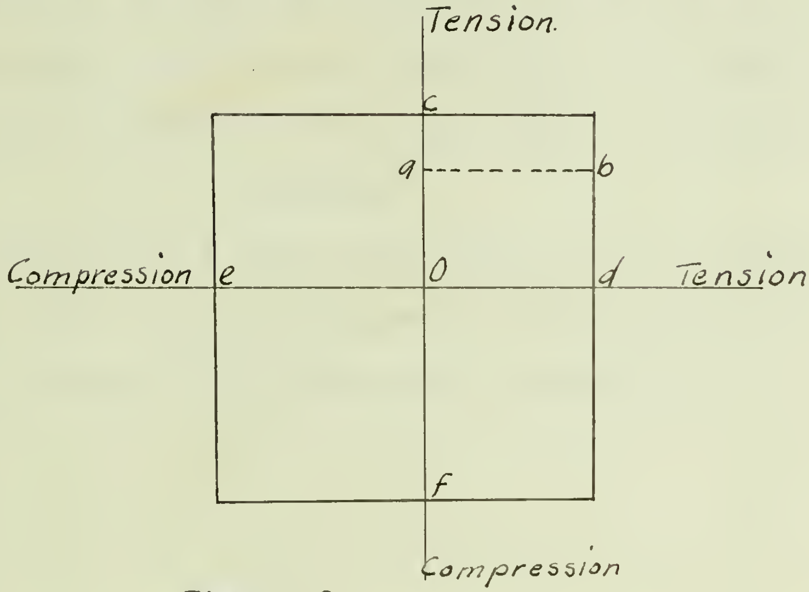

Figure 6.

This diagram is to be used somewhat differently than the preceding. If a tensile stress ab (Figure 6) is applied in one direction, any stress (oa) may be applied at right angles, provided oa does not exceed oc = ab = the yield-point stress. This is on account of the assumption that a stress is not affected by a second stress at right angles to it.

Maximum Shear Theory.

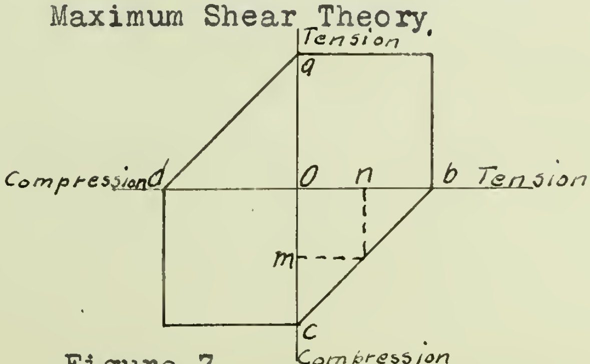

Figure 7

The first and third quadrants (Figure 7) correspond to the maximum stress theory, so that for stresses of like signs, one stress is not affected by the application of a second stress at right angles. Yielding will take place when either stress reaches the yield-point stress. For a combination of tension and compression (second and fourth quadrants) the lines cb and ad are inclined, because the tension plus the compression is a constant and is equal to twice the shear. (See page 23)

$$t + c = \text{constant}$$

By setting t and c each equal to zero in turn, it is seen that t must equal c, and this theory demands an equal yield-point stress for tension and compression.

PART III

EXPERIMENTAL WORK

Part III

Experimental Work

1. Form of Specimen.

The determination of the type of specimen to be used in the experimental work was a problem of considerable magnitude. Specimens subjected to direct tension or compression were not considered on account of the method of application of the load. A cube subjected to compression in two directions could easily have been set up, but the friction between the bearing blocks and surfaces of the cube introduces inequalities and resistance to the change in cross section which could easily vitiate the results. (See Zeitschrift des Vereins Deutscher Ingenieure, 1900, page 1530 et seq.) Also a large number of short square steel bars, closely spaced to form in effect a plate, were not considered to sufficiently obviate this difficulty. Similarly, a tension specimen held at the four edges would not give good results. Direct stress application seemed out of the question, and recourse was had to bending to produce stresses in two directions at right angles to each other.

The first bi-axial stress experiments in this series of tests were made upon flat cross shaped specimens subjected to cross bending to produce two compressions or two tensions at right angles to each other. The stress distribution was so far from regular that no safe comparisions could be made.

The difficulties encountered in the cross bending tests

led to the adoption of tubes for test pieces on account of the facility with which bi-axial stress could be applied by means of an axial pull in a testing machine and internal hydrostatic pressure to produce a circumferential tension, and because the thinness of the wall and the relatively large tube diameter made the stresses practically uniform throughout the tube. The stress-strain diagram should show a much sharper break than is the case where solid bars are used as specimens and the yield point is more positively determined. There are no greater eccentricities of application of load when using the tube than when working with a solid bar, and on account of the greater diameter of the tube, this eccentricity is relatively less important.

Objection has been raised by several investigators* to the use of tubes on account of the difficulty of accurately measuring the tube thickness and because instruments cannot be attached to the tubes. These things were urged as reasons for lack of confidence in the results.

The method adopted for measuring the thickness of the tube walls is thought to be accurate and the check results obtained by measuring the tube with a micrometer after it had been cut, have borne out this conclusion. The tube must be of relatively large diameter in order to apply this method, but with six inch tubes, no difficulty was experienced. Seamless drawn tubes are of fully as uniform quality of material as solid bars, while the uniformity of results shows that the material was quite uniform in quality. For a description of the tubes as tested, see p. 49 and 57.

* W. A. Scoble, Phil Mag., 1906
* C. A. M. Smith, Inst. Mech Engrs., 1909.

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Strains were measured by means of a Berry strain gage, using a four inch gage length. The accuracy and reliability of an instrument of this type has been demonstrated repeatedly and reference is made to the tests by A. N. Talbot and W. A. Slater on reinforced concrete buildings, as given in Bulletin #64 of the University of Illinois, to show what results may be achieved with such an instrument. A discussion of the strain gage and its use is given in a paper by Slater and Moore in the 1913 proceedings of the American Society for Testing Materials.

The use of the strain gage marks a decided advance in the measurement of strains. With this instrument it was possible to take twenty-eight separate readings for each increment of load, whereas other investigators have been able to take four at the most, and often only two. The advantage of a portable instrument over an attached one is very great and the rapidity of operation and freedom from danger by jarring the instrument, as well as the ability to read overlapping gage lines is a decided step in advance.

The total number of readings taken approximates 4300 on the tubes alone, while the uniformity of the results attests the confidence that may be placed in them.

The greatest disadvantage encountered in the preparation of the test pieces was in the slight change of shape of the tubes after machining, especially after the boring. This renders the tube slightly elliptical (but not over .02inches in 5.50 inches.) and of varying thickness. While this variation in the thickness was as high as fifteen per cent in some instances, it apparently did not affect the averages of the readings, although the individual

[illegible]

circumferential curves show such an effect and the effect of the water pressure in making the tube truly cylindrical.

It is much easier to cover the total range of stress ratios by the use of hydrostatic pressure and axial tension or compression in the tubes, than to use torque and axial load on solid bars. The latter method is inferior to the tube tests since only a small portion of the material is carried to the yield-point stress. The experiments are more successful when as much of the specimen as possible is uniformly stressed, and the best condition is that wherein the entire specimen is uniformly stressed. This is true both on account of the yield-point effect and on account of the smallness of the strains to be measured. The use of tubes of reasonably large diameter seems the best method for investigating bi-axial stress.

2. Cross bending Tests.

The description of the experimental work done in crossbending and the tables of the date are given for two reasons, first to make a record of the work done, and secondly, should any one later on wish to investigate the subject--or this data in particular--it will be available.

Specimens for the cross bending tests were prepared from one-fourth inch soft steel plate of the shape and dimensions shown in Figures 8 and 10. Two pieces were tested and strips from the portions cut away were tested in tension.

Series 1.

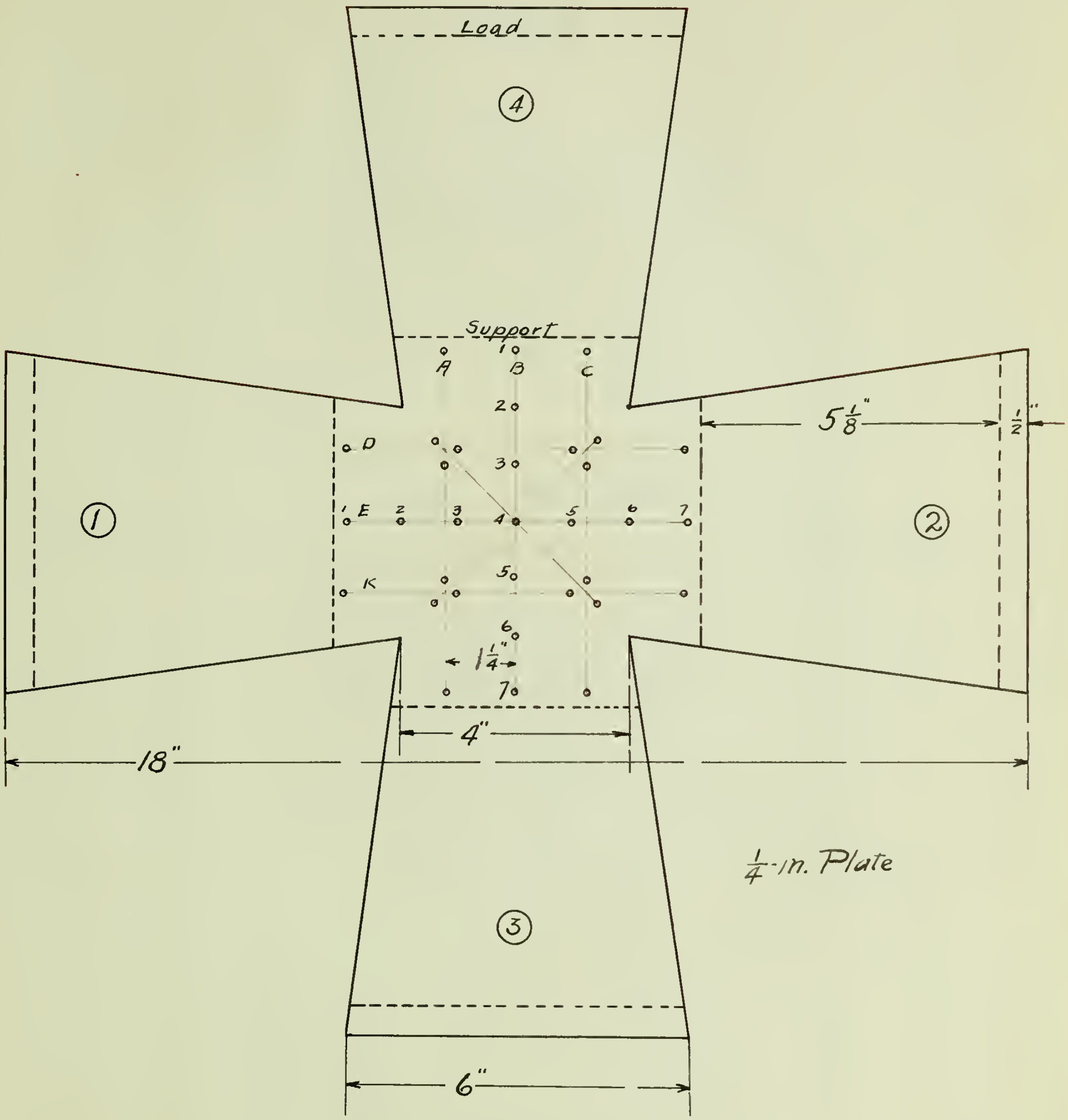


Fig. 8.

Crossbending Specimen.

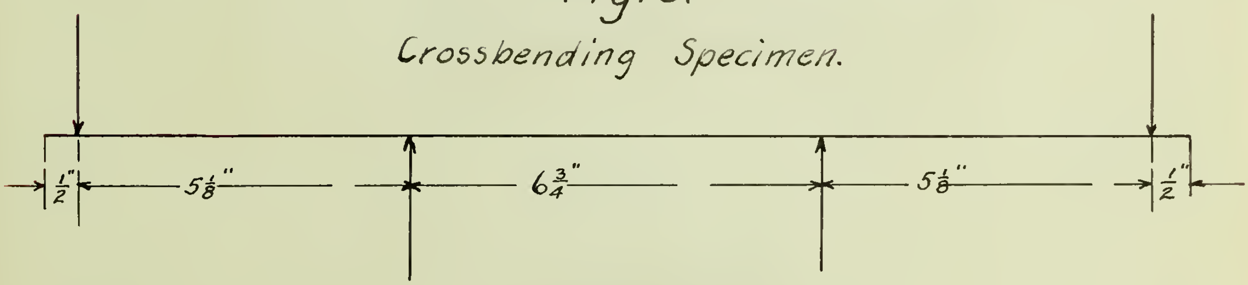


Fig. 9.

Series 2.

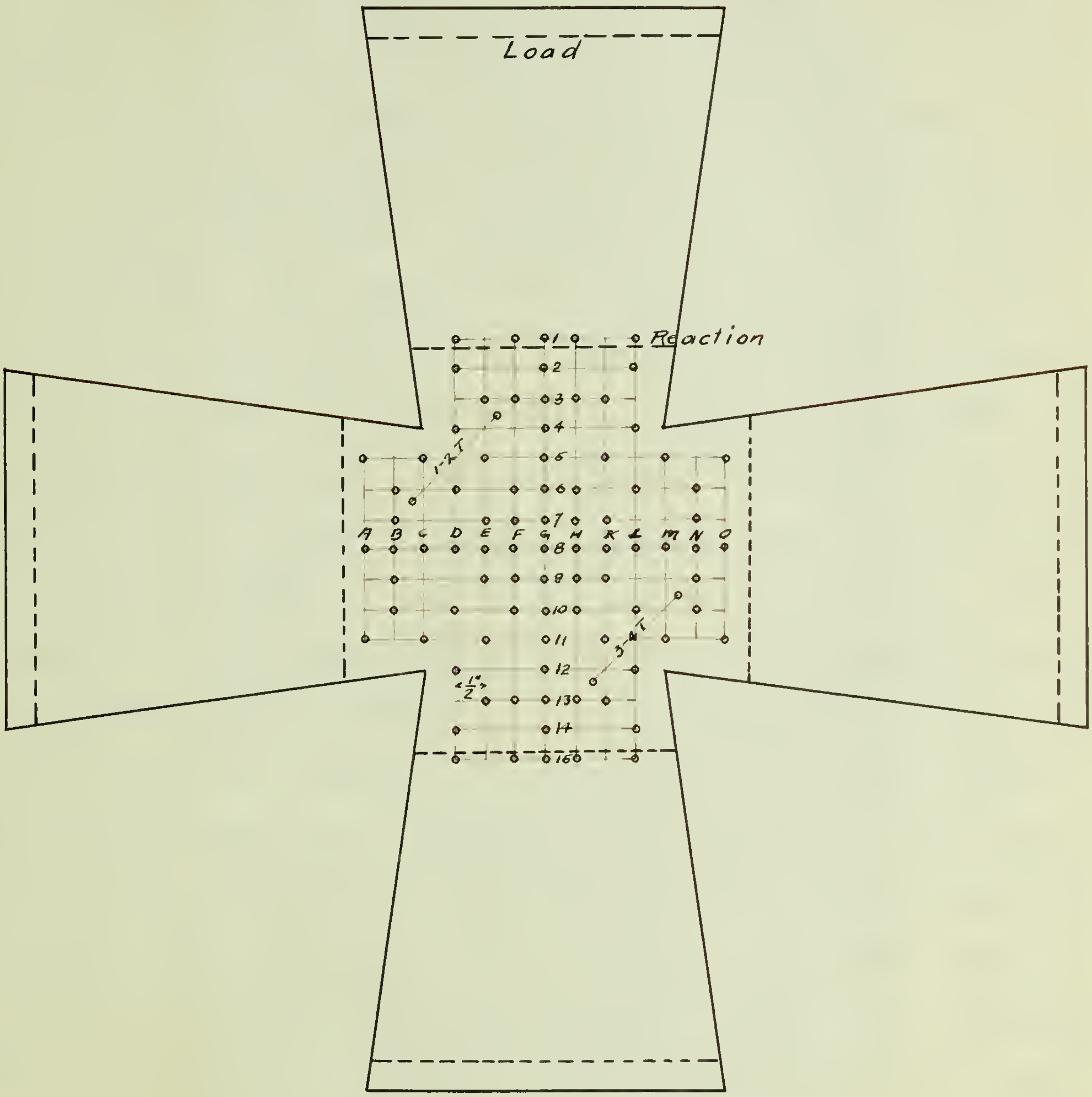


Fig. 10.

Crossbending Specimen.

The set-up for the bending tests is shown in Figures 11 and 12. In order to have the upper surface unobstructed for the use of the strain gage, the beam was loaded as an overhung beam with four symmetrical loads, two each in two directions at right angles to each other. The loading diagram in one direction is shown in Figure 9. The center part of the cross was subject on the top to two tensions at right angles to each other.

Load was applied by placing known weights on the yokes at the ends of the arms of the specimen, thus giving a definite bending moment. The strains were measured over two-inch gage lines with a Berry strain gage. Instead of a uniform stress over the center portion of the test piece, the readings showed a considerable variation. The yield point was reached first at the corners, and the lines of yielding spread inward, curving toward the adjacent corners. As the load was increased, these lines increased in number and others appeared just outside the center of the cross. These latter lines were straight and were parallel to the line of the support. The effect of the sharp reentrant angle at the corner in changing the lines of stress must have been considerable, for failure started along a line making approximately 45 degrees with the center lines. The line from each corner divided and each line gradually changed direction becoming parallel to the lines of symmetry of the specimen, until the lines from adjacent corners joined. The lines of yielding are clearly shown in Figure 13, which is three quarters size. The square from corner to corner and the center lines were used to lay out the specimen, and must not be confused with the yield lines. It will

Fig. 11.

Fig. 12.

be observed that the specimen, considered as a beam, widens suddenly for the center four inches, but the effect of this increased width in carrying stress was slight.

Stress-strain curves along three lines (the center line and two outside lines) in each direction for Series 2 are given in Figures 23 to 28 inclusive on pages 102 to 107 . All four arms were loaded. The stress-strain diagrams for twelve of the gage lines showing the greatest elongations are given in Figures 29 and 30 on pages 108 to 109. The piling up of the stress at the corners and the variation from point to point is well shown by these curves. The place of greatest stress is evidently a small area near each corner and to get the maximum stress would require a very short gage line. This stress condition is an indication of a condition due to the form of the specimen rather than to a condition of combined stress.

No further use was made of the data of these tests (See Tables I to VI) because the non-uniformity of stress distribution rendered a comparison with tension tests uncertain and any conclusions drawn from such a comparison would be misleading. Table VII gives the applied stresses calculated on the basis of the bending moment, neglecting any increase in width of section at the middle. The computation was based upon the usual beam formula and takes no cognizance of the influence of another stress at right angles.

3. Preliminary Tube Tests.

The failure of the bending tests to give definite results led to the adoption of tubes which were to be subjected to axial compression or tension and hydrostatic pressure to produce hoop

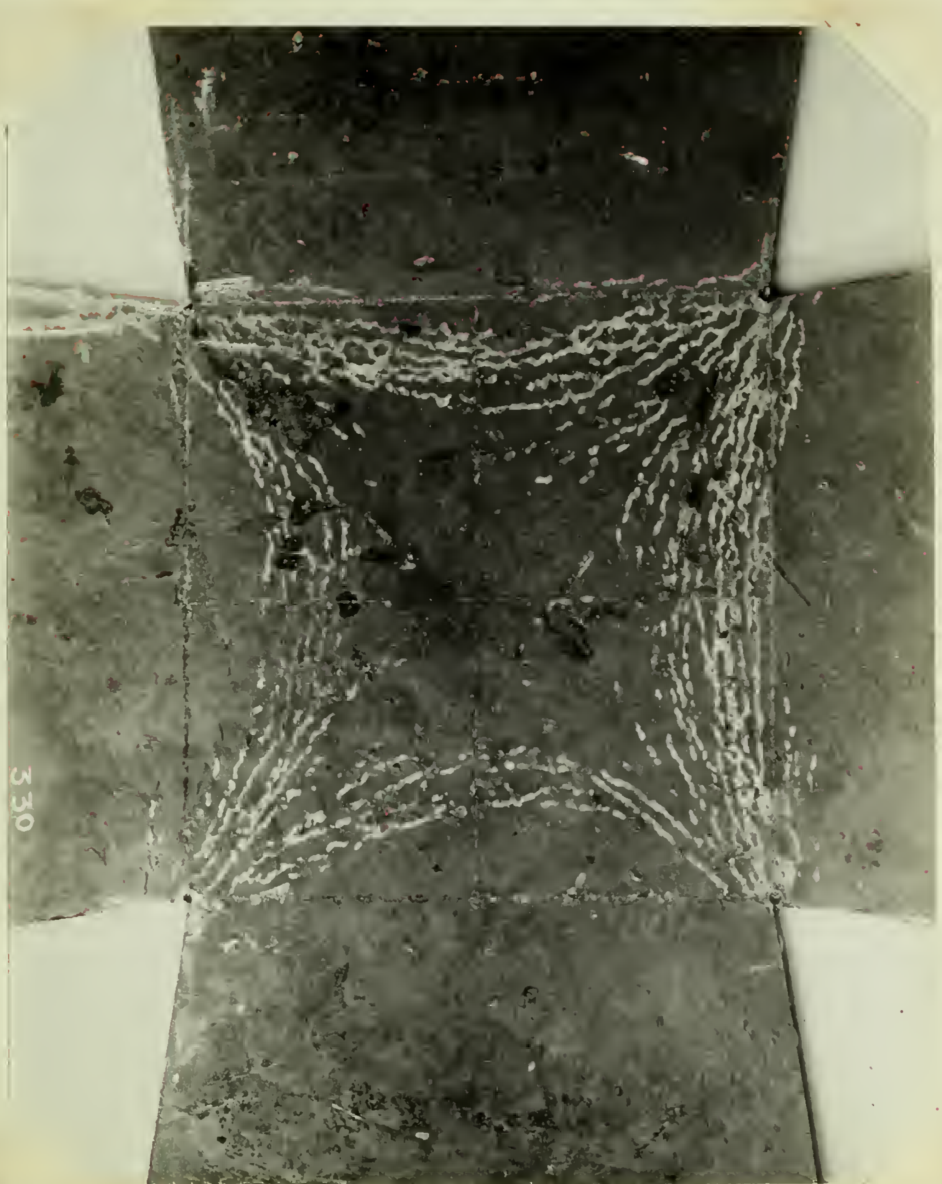


Fig. 13.

tension. This method gives two principal stresses at right angles to each other, the third stress being small since it varies from the intensity of the hydrostatic pressure on the inside to zero on the outside. It had been hoped to use tubes about twelve inches in diameter, but the cost was prohibitive, so a compromise selection resulted in the adoption of 6 inch tubes with $\frac{1}{4}$ inch walls. The thickness of the tubes when finished varied between .082 inches and .092 inches, as the average thickness at a given cross section.

To determine the value of this type of specimen, a single tube was prepared, 2 feet 2 inches long by 5.50 inches internal diameter. The tube was threaded on the two ends with a taper thread of twelve threads per inch over a length of 3 inches, and the ends faced. The remainder of the tube was turned down to an approximate thickness of 3/32 inch, leaving four bands of $\frac{1}{4}$ inch width having a pitch of four inches, symmetrically placed along the tube. The greater part of these bands were afterward milled off leaving four projections on each band for the gage holes. It was planned to span the tube with four gage lines of four inches each. In order to make proper provision for the gage holes the projections on the tube were milled as shown in the accompanying sketch, Figure 14, and in the photograph, Figure 14a.

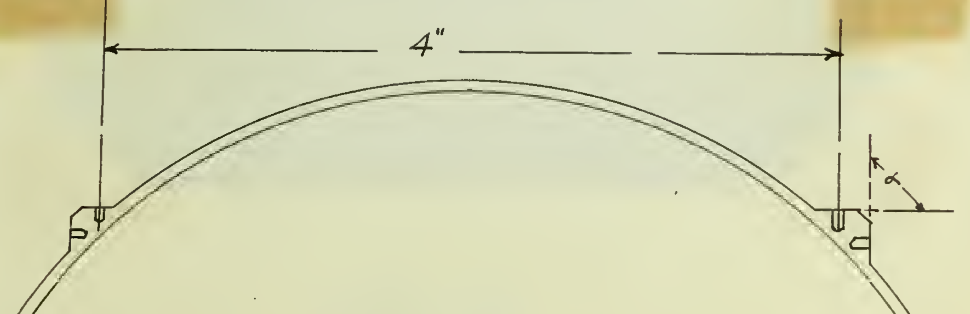


Figure 14

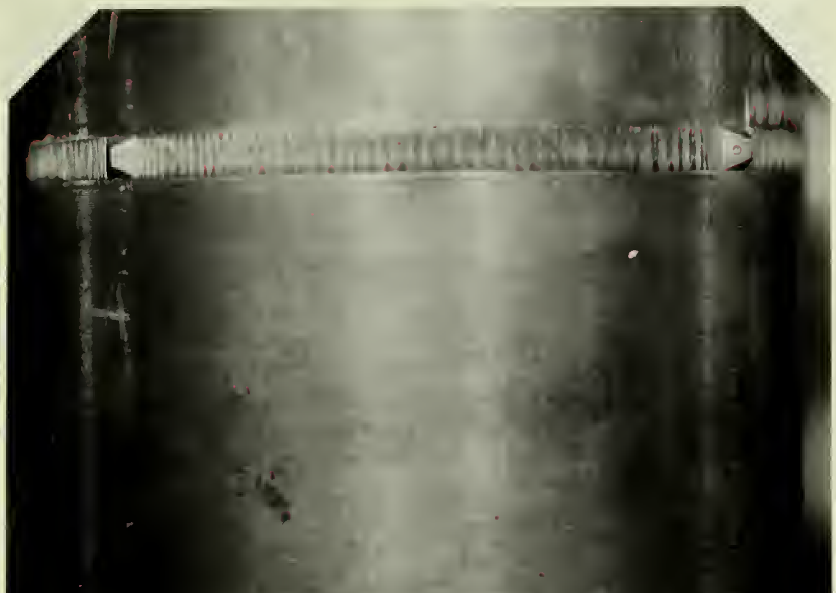


Fig. 14a.

Fig 14b.

Top.

Bottom

Developed Surface of a Tube.

As there were four projections around the circumference, the angle α was 90 degrees in order that the gage holes in two consecutive projections would have the same surface plane. The axial gage lines used one of the two holes so that the projections could be reduced to the smallest possible size. This gave four rews of three axial gage lines each, twelve in all, and four bands of four circumferential gage lines, sixteen in all, making it necessary to take twenty eight readings, exclusive of the standard bar and check readings, which are necessary in tests with the strain gage, for each increment of load. The circumferential bands were lettered A, B, C, and D; the axial lines of projection were numbered 1, 2, 3, and 4. Thus an axial gage line would take two holes in the same axial line, but in two consecutive circumferential bands. It would, consequently, be called by the letters of the bands in order, and by the number of the axial line. Thus BC1 would be an axial gage line spanning the distance between the circumferential bands B and C and lying along the axial line 1. (See the photograph, Figure 14a and the developed surface of the tube Figure 14b.)

As soon as the tube was machined, the numbering was determined and the projections on the A band marked with small prick punch marks to identify the axial lines. In this way the readings for the thickness of the tube walls could be correlated with the strain gage readings.

4. Determination of the Thickness of Tube Walls.

The principal of the apparatus adopted for measuring the thickness of the tube walls is that of a micrometer caliper

with a very deep throat. The first device used is illustrated in Figure 15. It consists of a stiff wooden bar clamped to a 4-inch I beam and carrying an Ames dial reading to one thousandths of an inch. The plunger of the dial rests on a steel ball embedded in plaster paris and set in a wooden strip securely fastened to the I beam. The zero reading is then taken with the plunger resting on the highest point of the steel ball. The I beam is sufficiently stiff so that the weight of the tube does not deflect it measurably. To determine the thickness of the tube wall the plunger of the dial is raised, the tube is slipped over the I beam and wooden strip and rests on the steel ball. When the plunger is in contact with the tube, the thickness is the difference between the reading then taken and the zero reading. Two other steel balls are embedded in the wooden strip on the I beam, one on each side of the ball under the plunger, at such a distance from it that the tube always swings free on the center ball and one of the others. The center ball--the one under the plunger of the dial--was slightly higher than either of the others to insure a bearing on it at all times. By slipping the tube along, readings could be taken rapidly. Zero readings were taken after a traverse of each axial line. A set of check readings was taken and where there was any appreciable variation, the average was used. The dial was read to tenths of a division (ten thousandths of an inch) and the check readings were never more than one one-thousandths of an inch at variance with the original set.

As the later tubes were longer, the over hang of the beam required to measure the thickness was so great that the variation in the deflection due to the shift of the tube became

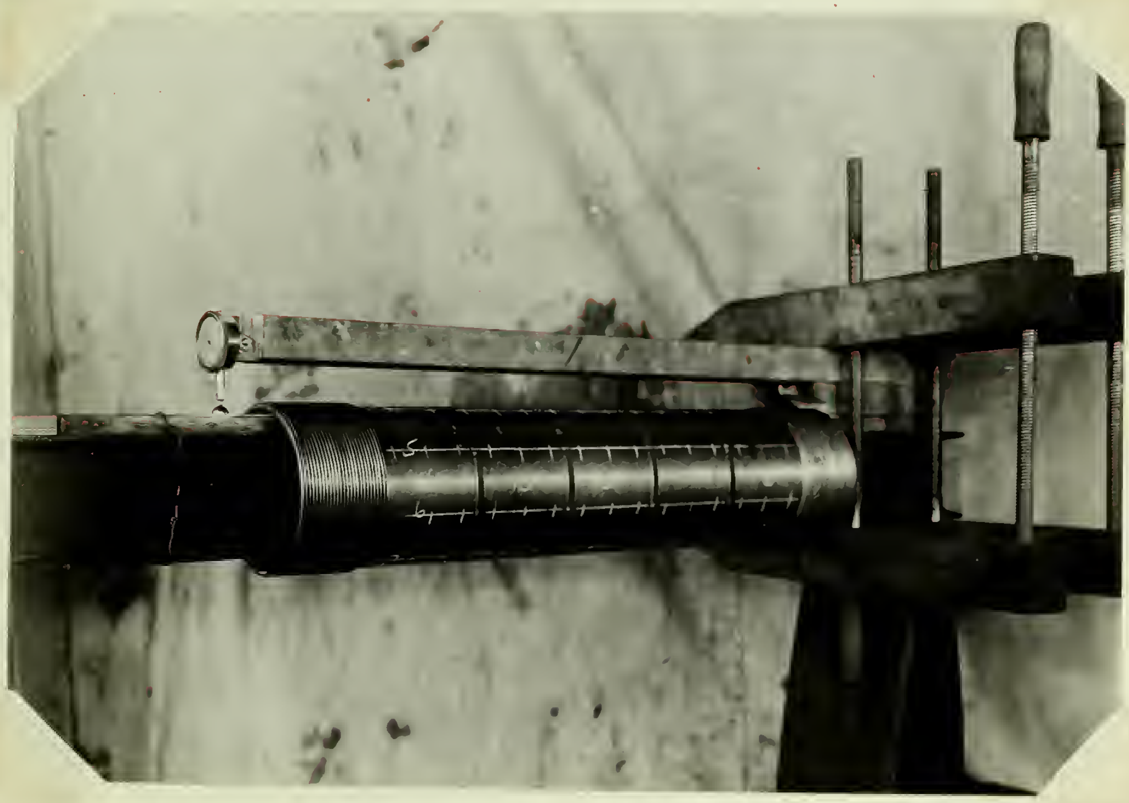


Fig. 15

noticeable. The apparatus was changed slightly and a $4\frac{1}{2}$ x$2\frac{1}{2}$ x 7/16 inch T bar was clamped to a support at one end and the wooden bar carrying the dial was bolted to it. Readings were taken as before, but the zero readings were obtained by suspending the tube in two fine wire slings in such a manner that its weight came on to the T bar in the same way as when it swung on the steel balls. Then with the plunger of the dial resting on the central ball--which in this case was set into a hole drilled in the stem of the T bar itself--the initial or zero reading was taken for every position of the tube along an axial line. This gave slightly different zero readings for the various positions of the tube, but it removed any error arising by virtue of the deflection of the apparatus.

Figure 16 shows the T bar device and the method of suspending the tube for zero readings.

Measurements were taken every two inches along the axial gage lines and on lines half way between these. The latter lines have been marked 1a, 2a, etc. 1a lies between lines 1 and 2, 2a between lines 2 and 3. The thickness of all the tubes is given in Tables VIII to XVI, from which not only the thickness can be found, but the variation in the thickness as well.

5. Preparation of the Later Tubes.

After the first set of thickness readings had been taken it was found that the walls of the tube were not uniform enough to warrant making a test of the tube. The tube was then taken to the machine shops and bored out on the horizontal boring mill, which greatly reduced the variation in thickness.

Two Shelby seamless drawn tubes were bought in the open

Fig. 16.

market and made into test specimens. Five specimens were cut from one tube, numbered 1 to 5 inclusive; these constitute Series 1. Four specimens were cut from the other tube, numbered 6 to 9 inclusive, which constitute Series 2. There was some difference in the physical properties of the material of the two tubes, as may be seen from Figure 31 which gives the average stress-strain diagrams of the tensile tests on specimens cut from the tubes. These specimens were first bored out for the entire length and afterward turned down to the dimensions shown in Figure 17. They were not annealed.

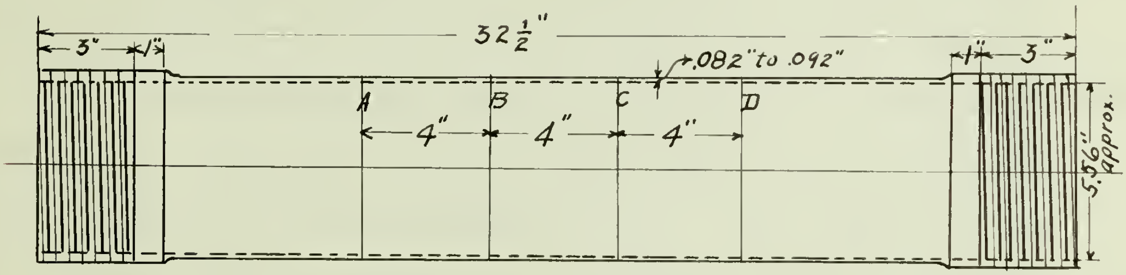


Figure 17

The projections for the gage holes were prepared in the same manner as for the first tube. The gage holes were drilled by hand, using a No. 54 drill, which is the size that is always used in the laboratory for gage holes.

The boring of the tube caused a very slight change of shape of the walls due to the removal of the inner skin of metal, and after the outside was turned the thickness was uniformly varying, usually having two points of maximum thickness diametrically opposite and at 90 degrees from these two points of minimum thickness. This slight change in shape was more noticeable in some tubes than in others and made it impossible to secure uniform thickness without further careful work.

6. Method of Testing.

Two steel castings were designed to fit over the ends of a tube. (See Figure 19). Material was ordered with an ultimate strength of 80,000 pounds per square inch, but the stresses the heads had to carry were comparatively low, for the maximum load was but 102,100 pounds, and the material was about $\frac{3}{4}$ inch thick at the thinnest part. The castings were machined all over and threaded internally, at one end to receive the tube and at the other to receive a $4\frac{1}{2}$ inch bar which served to apply the tension. The two threaded portions were separated by about an inch of metal to retain the water under pressure in the tube.

The preliminary test had shown that it was necessary to provide something more than a long fine thread to withstand the water pressure it was intended to use. The length of the thread (3 inches) was not sufficient to prevent leaks, even though the threads were carefully cut in a lathe. The collapse of the tube when it was removed from the mandrel tended to exaggerate this. Two layers of 3/8 inch hydraulic packing were used in an ordinary four screw stuffing box. The heads were recessed to receive the packing and the gland, while the tube walls were left nearly full thickness for an inch beyond the threads to furnish a firm bearing for the packing. After the packing was adjusted to position there were no perceptible leaks although pressures up to 1800 pounds per square inch were used. Figure 18 shows the general scheme of the first test, and Figure 19 shows the heads with the packing in position.

Fig. 18.

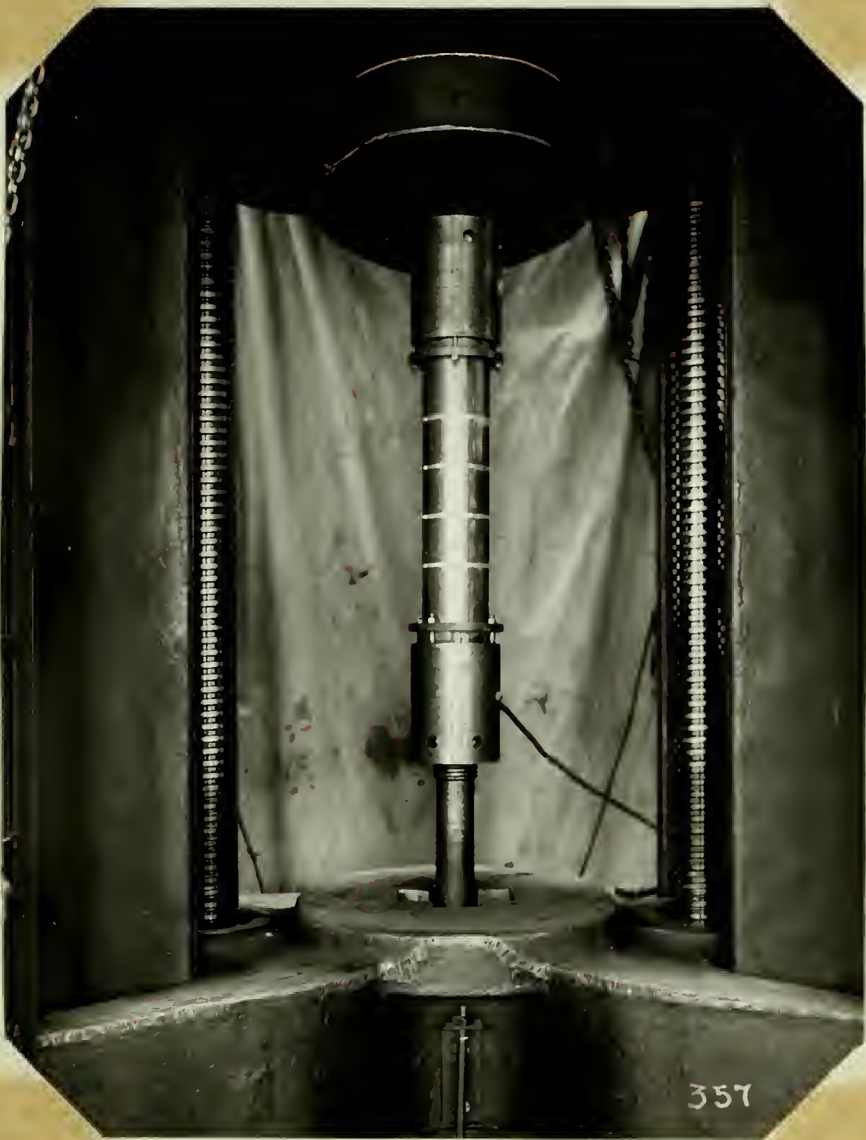


Fig. 19.

All the tests were made in the 600,000 pound Riehlé machine of the Laboratory of Applied Mechanics of the University of Illinois. Two spherical seats were used, the lower one inverted, with the nuts of the $4\frac{1}{2}$ inch bars bearing against them. In this way, with careful centering of the specimen in the machine, the eccentricity of loading was reduced to a minimum and the bending stresses were low. The length of thread on the specimens tended to give a good distribution of load and the distance of the first gage hole from the end of the thin part of the wall ($6\frac{1}{2}$ inches) together with the thinness of the wall itself, were sufficient to insure a high degree of uniformity of stress.

Holes were drilled into each head to connect into the interior of the tube, the hole in the lower head for connection to the pump and the hole in the upper head for the purpose of filling the tube with water. Each hole was tapped with a $\frac{1}{2}$-inch pipe tap.

During the first two tests a small jack (shown at A in Figure 20) was placed in a 100,000 testing machine and filled by the pump shown at B. The operator could observe the gage on the pump while running down the testing machine crosshead to keep up the pressure while readings were being taken. A needle valve was also provided, but was not used. The efficiency of the packing used in the later tests rendered this auxiliary accumulator unnecessary. It was not connected up after the second test, but instead the pump was connected direct to the lower head of the specimen.

7. Character and Sequence of the Tests.

As planned, the program of tests included for each series

[illegible]

Fig. 20.

a test in direct tension only, a test with the hoop tension one half the axial tension, and a test with the ratio of the hoop tension to the axial tension equal to .94. The ratio of .94 was adopted as the maximum rather than 1.00 because the difference in the strength of the steel in the two directions, with and across the direction of drawing, would possibly complicate the problem, if the higher ratio was used, and it was not intended to raise the question of the variation in strength in different directions throughout the specimen.

The remainder of the tubes were to be reserved until the data obtained from these tests had been worked up. This program was changed somewhat for Series 2 because the very low yield-point stress developed caused the loss of one tube, as the yield-point stress was passed before a sufficient number of readings had been taken to establish it and the stress-strain diagram. The test in which the stress ratio was .94, of this second series, was not wholly satisfactory and the fourth tube was used to check the results of this test. Specimens cut from the parts of the tube that were not heavily stressed were tested to obtain the strength in axial tension. (See Tables XVIII and XVIIIA)

In Series 1, the last two tubes were tested at a ratio of hoop tension to axial tension of .24 and .69 respectively. The axial stress on a tube was computed by adding the load registered by the testing machine to the product of the area of the inside of the tube in square inches and the water pressure in pounds per square inch. Thus, supposing a ratio of hoop tension to axial tension of .50 were used, since the axial tension due to the water pressure is one half the hoop tension produced by

this internal pressure, the total axial load would have to be four times the axial load due to the water pressure. As the average area of the inner cross section of the tubes was about 24.50 square inches, this axial load was 2450 pounds per 100 pounds per square inch water pressure. To produce a ratio of hoop tension to axial tension equal to .50, requires a machine load per 100 pounds per square inch water pressure of

4 x 2450 − 1 x 2450 = 7350 pounds, since the water pressure acts with the machine load. Then dividing the total load (4 x 2450 pounds) by the area of the tube, the unit axial stress is obtained. The hoop tension is one half of this value in this case. A slight error is introduced by using the inside diameter of the tube rather than the mean diameter, for, in order that the hoop tension shall be exactly twice the axial tension, when the water pressure only is acting, the mean diameter must be used. This error is only about $1\frac{1}{2}\%$.

Series 1 in full is as follows:

Tube Number	Ratio of hoop tension to axial tension	
	No gage correction	After correcting for gage reading
5	Axial tension only	——
1	.25	.24
2	.50	.475
4	.70	.69
3	.94	.92

Series 2

Tube Number	Ratio of hoop tension to axial tension No gage correction	After correction for gage reading
7	.50	.475
8	.94	.92
6	.94	.92
9	Axial tension only--no result	

The pump is a Watson-Stillman pump which is used to supply oil under pressure to hydraulic jacks. It is part of the regular laboratory equipment of the Laboratory of Applied Mechanics of the University of Illinois. The gages used were Crosby hydraulic gages reading to 1000 and 2000 pounds respectively. They were calibrated on the 1500 pound Crosby gage tester in the Mechanical Engineering Laboratory and the 1000 pound gage was checked on the Crosby dead weight tester of the Laboratory of Applied Mechanics. The results were identical. This calibration may be found in Table XVII.

The strain gage used is a 4 inch Berry made for the Joint Committee on Stresses in Railway Track, and loaned by that Committee. It has invar steel sides and shows a negligible correction for temperature. Two standard bars were used to catch any variation of the instrument due to jarring or striking the fixed point. All tabulated data has been corrected for variation in the standard bar readings. (See pages 158 to 173) To avoid any variations due to temperature the tubes were usually filled with water in the evening, and by the time the test began the next day the tube and the water were at a temperature that scarcely changed during the entire test.

8. Test Operations.

The initial load in all cases was small, producing an average unit stress of approximately 4000 pounds per square inch. This load was applied after the specimen had been carefully centered and the spherical seats tried. In all tests except those in axial tension only, three persons were required to carry on the work, instrument man, recorder, and pump operator. Computations were made in advance to determine the loads required, the approximate yield point, the water pressure, machine load, and unit stress. This program was carried out as planned unless the elongations obtained showed too great divergence from the average. In such cases smaller increments were taken. When the load was increased, the water pressure was increased first and then the machine load. The load increments were small enough so that the stress ratio was practically constant at all times.

The record of a test was a combination of the ordinary record and a graphical one. Figure 21 is a photograph of one of the record sheets, and is typical. Coordinate paper was used and was divided up into a series of rectangles, one for each standard bar and gage line. Along one side of this rectangle the instrument reading was noted and the reading then plotted. In this way the progress of the test was very evident and any reading not conforming to what seemed right was checked to insure its correctness. It may be contended that the data should be taken blindly, without any knowledge of the character of the results. In some cases this may be true, as in a chemical analysis, or where the law is not known, but where the nature of

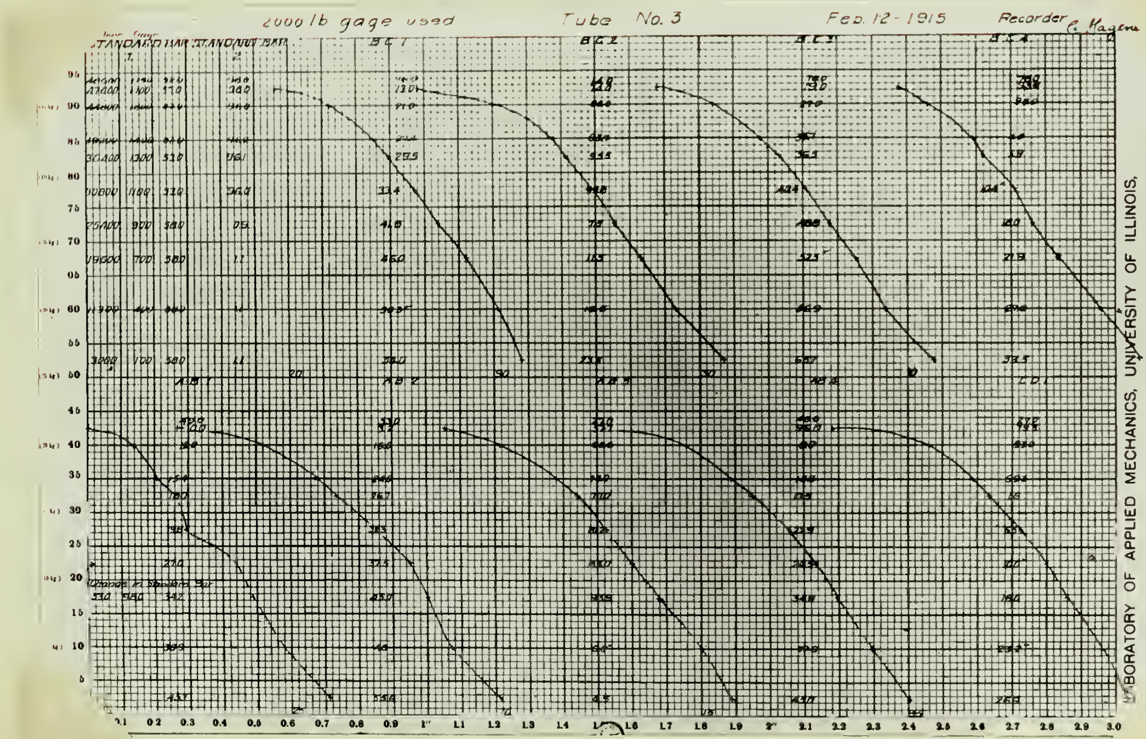

2000 lb gage used
Tube No. 3
Feb. 12 - 1915
Recorder
STANDARD BAR
STANDARD BAR
LABORATORY OF APPLIED MECHANICS, UNIVERSITY OF ILLINOIS.
Laboratory No. of Test
Test of

Fig. 21

the curve is well known, it is necessary to see that the results that do not conform to this law are checked to insure their accuracy. If this is not done, false breaks may sometimes be obtained in the curve, and it is much better to check the reading than to get a false result. If the error is experimental, the check reading will correct it, and if the stress suddenly departs from the straight line law, the check reading will be a repetition of the first reading and will give greater confidence in the result. Even if only one or two errors are discovered and corrected, the result justifies the method employed. Whenever there are variations from the straight line in the stress-strain diagram, these are indications of a change in the rate of taking stress. As the load changes, the distribution of stress over a given cross section often changes, so that at one point there may be a rapid increase in the elongations for one increment of load, while in the adjoining gage line the change is slight. The next increment of load may bring about a complete reversal of the conditions shown by the previous instrument readings. Whatever variation occurs in one gage line, it is reflected in one or more of the others, so that the average takes out all these peculiarities. This is especially true of the circumferential readings.

Circumferential gage line readings give the correct unit elongation when the chord length is used instead of the arc-length, for while the total elongation is greater along the arc, the total distance between the gage holes is greater, and, since the subtended angle (90 degrees) always remains the same, the unit elongation is correctly obtained from the chord length.

Circumferential readings are subject to the tendency of the tube to become truly cylindrical under water pressure. This varies with the ratio of the hoop tension to the axial tension and depends upon the water pressure. Where high water pressure is used, this disturbing effect is not so noticeable as the test progresses, while for low pressure the effect is sufficient in some cases to change the stress from a tension to a compression, or vice versa.

Axial gage lines were read first, all AB lines first, in order, then the BC lines, and lastly the CD lines (See Figure 14b) The circumferential lines were read in the following order, 1-2-A, 1-2-B, 1-2-C, 1-2-D, 2-3-A, 2-3-B, 2-3-C, 2-3-D, 3-4-A, 3-4-B, 3-4-C, 3-4-D, 4-1-A, 4-1-B, 4-1-C, 4-1-D. This is merely a matter of personal convenience to the instrument man. Standard bar readings were taken before and after each set of readings. In all the tests a total of 6200 strain gage readings, exclusive of check readings, and 2100 tube thickness readings were taken. 330 extensometer readings were taken when testing the specimens in direct tension.

9. Diagrams and Tables.

Stress strain diagrams of the average results of a cross section are given on pages 111 to 119 for the axial lines, and on pages 120 to 128 for the circumferential lines. The general average diagrams for the axial lines are given on pages 129 and 130, and for the circumferential lines on pages 131 and 132.

Tables of the data of the cross bending tests are given on pages 136 to 144 in Tables I to VII inclusive, and that of the tube tests on pages 145 to 173 in Tables VIII to XXXIX.

The theoretical and experimental results are shown graphically on page 131, and comparisons of the different theories are made on pages 132 and 133.

PART IV

DISCUSSION OF RESULTS

PART IV

Discussion of Results.

1. The Criterion of Strength.

1. Limit of Proportionality.

There are three possible stress limits which may be selected as the criterion of the strength of material, limit of proportionality, yield point, and rupture. It is recognized that there may be a sharp distinction between the laws governing ductile and brittle materials, for such a distinction is observed in the stress-strain diagrams and in compression and torsion failures. Since this discussion is limited to ductile materials, conditions will be treated only as they apply to such materials.

It would appear at the first thought that the limit of proportionality would be the proper basis upon which to determine the relative strength of material. The mathematical theory of elasticity is based upon Hooke's law generalized, engineering practice bases it computations largely upon this same law, and several investigators have used the limit of proportionality--which they incorrectly called the elastic limit--as their criterion, notably Hancock and Turner.

The limit of proportionality, or p limit, is defined as the stress at which the constancy of the ratio of stress to strain ceases; that is, the modulus of elasticity is a constant up to this stress, and beyond it permanent set takes place. It is often stated that the distinction between yield point and p limit is very slight, that it really makes no material difference which is used. But a glance at the stress-strain diagrams on pages111-118

will show that in some cases the modulus of elasticity changes and that the diagram consists of a broken line instead of a straight line nearly up to the yield point. This fact, due to the lack of isotropy in the material and to the mechanical work done upon it, makes it difficult to get consistent results by using the p limit as a criterion. When the material has been cold worked, the stress-strain diagram often curves away from a straight line slowly and the exact point of departure is not easily located. Special treatment of the material usually affects the yield point in the same way in different specimens, but not the p limit.

2. Rupture.

The use of rupture, as a criterion of the strength of ductile materials, still persists in the simple stresses, for specifications ordinarily require that the ultimate strength of the material shall have a certain value. But this is an indirect measure of the toughness rather than of the strength, and in the best specifications the yield point (or elastic limit, as it is frequently but incorrectly called there) is specified as well. Conditions at rupture give no indication of those existing at the yield point and whatever value a knowledge of the conditions attending rupture in a ductile material may have, no conclusions can be drawn from them which may safely be applied to the period preceding the yield point. As engineering design deals principally with stresses within the yield point, rupture cannot be considered as the criterion, even tho Bridgman* in his tests on thick cylinders uses it and decries the use of the yield point. When the distribution of stress is unknown and no extensometers

* Philosophical Magazine, July 1912 p. 63

are used to measure strain, rupture is the only criterion available but this will not justify its use for the purposes of this investigation.

3. Yield Point.

For materials that have not been worked cold, the stress-strain diagram shows a very decided change in character when the material passes from the elastic to the plastic condition. Most elasticians pay no attention to the intermediate period, as it is difficult to handle mathematically and occupies a relatively small part in the diagram. When the material has been cold rolled or cold drawn, this yielding is more gradual and the curve, instead of breaking sharply, is much slower in passing from one state to the other. If the specimen is tested in simple tension with an extensometer and the load is slowly but steadily applied, the roll of the curve is apparent a short time before the yield point is registered by the drop of the beam. If a time allowance has been made, near the yield point, for the particles to reach an equilibrium, this difference may be as great as 15%. The rapid increase in the strains after the yield point has been passed ordinarily puts an end to the usefulness of a structure. While there may be no utter collapse, it has to all intents and purposes failed on account of the relatively large permanent sets. An excellent illustration of this is given by the reinforcing bars of a reinforced concrete beam.

As all the investigations hereinafter described were made with instruments to measure the strains, some criterion must be adopted that is applicable to a stress-strain diagram. The first deviation from a straight line is an indefinite point

to locate and considering everything that has been noted here, the method proposed by the late J. B. Johnson has been adopted. This is called the "apparent elastic limit", although here it is taken as the yield point. This method empirically locates a point at which there is evidently some plastic action, and furnishes a very convenient method for comparison of results. It is defined as that unit stress at which "the rate of deformation is fifty per cent greater than it is at zero stress."

Figure 22 shows the application to a stress-strain diagram.

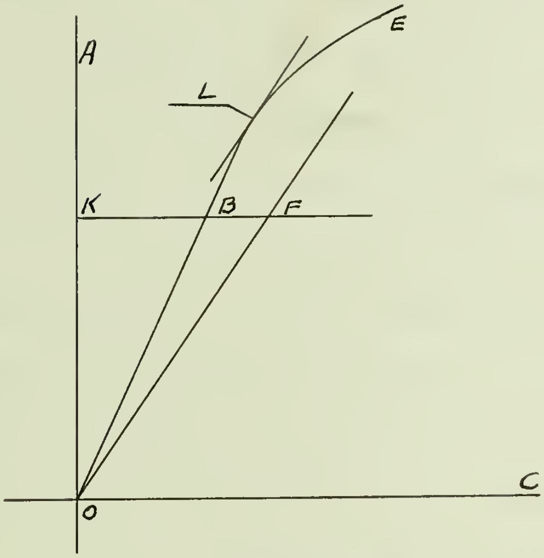


Figure 22.

Let OBE be a stress-strain diagram drawn in the usual manner. Then AOB is the angle determining the slope at zero stress. At any point K lay off horizontally a distance KF equal to 1.50 times KB. Then OF is the slope fifty per cent greater than zero slope. A parallel to OF drawn tangent to the curve BE, locates the point of tangency L and the corresponding stress is the yield-point stress.

2. Strength.

In the tabulation of the results of the tests of tubes

under bi-axial stress, the average of the elongations measured on the four gage lines at any one cross section (see Figures 14a and 14b) was taken as the elongation at that cross section. Thus, for the axial gage lines, AB1, AB2, AB3, and AB4, of a tube were averaged, and this average is given as the elongation of the AB lines of that tube. Likewise for the BC and CD lines. For the circumferential gage lines, 12A, 23A, 34A, and 41A were averaged; that is, the four gage lines which made a complete traverse of the circumference at any one cross section. There are three sets of results for the axial lines and four for the circumferential lines of each tube. These average results are given graphically on pages 111 to 126, and in tabular form in Tables XXIV to XXXIX. The curves of the average results were then used to obtain the general average results. The elongations were taken from the curves and the average of these values for the AB, BC, and CD lines of each tube gave the general average axial elongations, while the average of the A, B, C, and D lines, similarly obtained, gave the general average circumferential elongations. The only exception is in the case of tube number 1 where the average of the AB lines is omitted in the general average. The general average stress-strain diagrams are shown on the following pages:-

Series 1

Axial lines Figure 48, page 127

Circumferential lines Figure 50, page 129

Series 2

Axial lines Figure 49, page 128

Circumferential lines Figure 51, page 130

Each general average curve represents the average

results of twelve axial gage lines or of sixteen circumferential gage lines. Each curve has the tangent of Johnson's apparent elastic limit shown, the point of tangency is indicated and the corresponding stress is given.

An examination of the average curves shows that the yield-point stresses are quite uniform for the different sets of gage lines, and in close agreement with those of the general average curves. Because of this uniformity, the use of the general average curves as the basis of comparison seems justified. The circumferential elongations are plotted with the apparent hoop tension as ordinates, except in the case of tube number 5 where no internal pressure was applied. In this case the ordinates are the axial stresses, so that it is easy to determine Poisson's ratio.

If a diagram is drawn having the yield-point unit-stresses as ordinates, and the ratio of the hoop to axial tension as abscissae, a comparison can be made with the results demanded by the three theories. Referring to the diagrams, Figures 5, 6, and 7, on pages 35 and 37, it is seen that the maximum stress theory and the maximum shear theory demand that the yield-point stress shall be constant for all ratios. Mohr's theory and the internal friction theory have the same requirements; Wehage's theory demands a reduction in the yield-point stress, and the maximum theory demands an increase.

What is the law that governs? In the past, investigators have bound themselves to one law and have sought to determine it. By referring to the curves of general averages, (Figures 48 and 49) it will be seen that as the stress ratio increases the yield-point

stress rises unmistakably until the stress ratio reaches the value of approximately 0.50. After this stress ratio has been reached the yield-point stress remains constant, no matter what ratio, up to unity, may be used. If these yield-point stresses are plotted against tension ratio , (see diagrams page 131) the first part of the diagram is a line with a uniform rise and the second part is a horizontal line of equal yield-point stress.

If now the line of the maximum strain theory--which is the only one demanding an increase in the yield-point stress--is drawn, using the value of Poisson's ratio and yield-point stress determined from the tension test on tube number 5, it will be found that this line, starting from the yield-point stress in simple tension, fits the points determined by experiment up to a stress ratio of approximately 0.45. As the theoretical line continues straight, while the line through the experimental points breaks and becomes horizontal, it is evident that the maximum strain theory holds only to the point of the break. Remembering the duality of stress in the material, it is possible that tension ceases to be a governing factor and that the shear becomes dominant at the point where the break in the line occurs. In order that the shear shall be dominant, two things must occur. (1) The shearing stress must actually reach the shearing yield-point stress as determined by tests in pure shear, and (2) since the shear is one half the maximum principal stress, this maximum principal stress must remain constant. The first condition is important only in so far as the shearing yield-point stress must be greater than one half the tension yield-point stress. Otherwise the shear would be dominant at all times. This latter is the

contention of the maximum shear theory. Counting compression a negative tension and with the principal stresses numbered in the order of their magnitude, p_1, p_2, p_3, the criterion for shear is

Shear $= \frac{1}{2}(p-p_3)$, but as the third principal stress is zero, this reduces to $\frac{1}{2}p_1$.

It may be contended that the water pressure inside the tube constitutes a third principal stress (compressive) and that p_3 does not equal zero, but equals the intensity of the water pressure. This is not the case, for all the readings of the elongations were taken on the outside of the specimen where the third principal stress was undoubtedly zero, if the atmospheric pressure is neglected. Since all the determinations of yield-point stress were made where this third principal stress was zero, the effect of the water pressure as a third principal stress is not considered. (see page 21.) The maximum shear theory (Guest's Law) carried to its logical conclusion demands that the yield-point stress of the material subjected to two stresses of like sign shall not vary from that reached in simple tension, for the shear is the determining factor at all times. If this theory holds, a horizontal line drawn through the yield-point stress in simple tensions should pass through all the points. But experiments * have shown that for ductile material , such as was used in these experiments, the ratio of the shearing yield-point stress from torsion tests to the tension yield-point stress varies with the material, but usually lies between 0.55 and 0.65, in the majority of tests ranging near 0.60, which is the commonly

* L. B. Turner, Eng'g London, Feb. 5, 1909.

accepted value. A few tests show a ratio less than 0.50, but they are relatively small in number. With a ratio of 0.60, the shear yield line would lie above the line through the yield-point stress in simple tension by an amount equal to 0.20 of the latter stress. The exact location of the line will vary with the material, but whatever the percentage of increase in shear, there is the hiatus between this condition and that demanded by the maximum shear theory. Some law must govern between simple tension and the time when the shear controls. Several investigators* deny that this gap exists, but an examination of the tests they have published shows that it is very evident, and that they have endeavored to explain the variation by lack of homogeneity of the material, irregularities in the tests, and eccentricity of specimens. For further discussion of this topic see page 95a under Correlation of the Results of Other Investigators.

If the horizontal line through the experimental points is taken as the limit of the shearing strength, the ratio of shearing yield-point stress to tensile yield-point stress is .59 for Series 1, and .62 for Series 2, which values agree well with the majority of experiments.

These tests indicate that there is a dual law covering the case of combined stress, or rather two distinct laws, when the stresses are both tension and act in two directions at right angles. Apparently the point at which the break in the line occurs depends upon the ratio of the yield-point stress in shear to that in tension and the change from one law to the other may occur at different ratios of the principal stresses for different

* C.A.M.Smith, Inst. Mech.Eng.,1909
* L.B.Turner, Eng'g., London, Feb. 9, 1909

materials. This is shown by the two diagrams (Figure 52),for in Series 1 the yield-point stress for the tube having a stress ratio of .475 lies on the horizontal line, while for Series 2 the tube with the same stress ratio lies on the inclined line. It becomes of increasing importance to establish this ratio of yield-point stresses by means of careful investigations; for if this is not an approximate constant, the use of combined stress formulae will require a knowledge of such a ratio for all materials.

Tension tests specimens cut from the thicker part of the tubes, parts that had been stressed by lightly, were afterward tested to compare the results with those obtained from the tube in direct tension, and, if possible, to fill in the gap caused by the loss of tube number 9. The general averages of these tests -- approximately 20 in number -- are given in Tables XVIII and XVIIIA and graphically on page 110. The break of the average curve of the small specimens from tubes number 2 and 3 agrees closely with the break in the curve obtained from tube number 5 (axial load only), 42500 pounds per square inch and 43000 pounds per square inch respectively. It is thought that this result justified the use of the yield-point stress from the average curve for specimens from tubes number 6, 7, 8 of the second series. This value 21,500 pounds per square inch has been taken as the yield-point stress of the tube in simple tension for Series 2 and computations have been made upon this assumption, using the value of Poisson's ratio obtained from the Series 1.

Granting that this is true as a general law, the application of this principle of dual control to the other three quadrants or combinations of stress is very simple. The governing condition

is the ratio of the yield-point stresses in shear and tension. Taking this ratio as 0.60, a comparison of the more important theories is made in the diagram on page 132.

Since the internal friction theory has been shown repeatedly not to hold for ductile materials, and these experiments seem to have disproved Mohr's and Wehage's theories, no attention will be given them. Comparison is made only between the maximum stress, strain, and shear theories.

With the tensile and compressive yield-point stresses equal, the rectangle ABCD represents the maximum stress theory, the rhombus EFGH the maximum strain theory, and the figure LBKMFNL the maximum shear theory. The line LUPRK represents the dual control in the tension-tension quadrant, while KSTM represents it in the tension-compression quadrant. The lines UP and PR are parallel to the axes and at such a distance from them that the ordinate of UP and the abscissa of PR are each 1.20 times OL or OK. TS is parallel to KM and at such a distance from it that one half the sum of the ordinate and abscissa of any point between T and S is equal to .6 of OM or OK.

The construction of the other two quadrants is such that the figure is symmetrical about the bisectors of the quadrants. It is to be observed that the diagrams showing the comparison of theory and experiment (Figure 52) is but a portion of this diagram, corresponding to LUP in the tension-tension quadrant.

Another diagram (Figure 54) has sometimes been used in which the ordinates represent the shear due to torque and the abscissae represent the tension or compression due to axial load or bending. The shear is plotted to twice the scale of the tension or compression. If a circle with a radius OT, equal to the

tensile yield stress, is drawn with O as a center, it will represent the relation between shear and direct stress required by the maximum shear theory. It will be observed that OS, the shearing yield stress must therefore equal exactly one half OT. A circle with a radius equal to OQ, the shearing yield stress determined from torsion tests, is shown, and the St. Venant ellipse TBA beginning at T, the tensile yield-point stress. The dual law requires that the St. Vanant theory hold from T to B and that the shear shall govern from B to Q. This diagram is not drawn for four quadrants, because shear (torque) is applied only to tension or compression; the diagram is therefore drawn with tension to the right and compression to the left of OS. The two parts are symmetrical. Hancock's * ellipse has been added to show how closely he came to the results here advanced.

While none of the experiments here recorded have been made on specimens subjected to combined torsion and axial compression or tension, comparison will be made under Correlation of the Results of Other Investigators, with the results obtained by other investigators.

The net result of this investigation as it affects the strength of steel under combined stress in two directions, both stresses tension, and the chief point of the entire discussion is, that instead of a single law, whatever its nature, as has heretofore been assumed, there are two distinct laws governing the strength of materials under bi-axial loading, each law dominant within its limits.

These two laws are the maximum strain theory and the

* Am. Soc. Test. Mat. 1908

maximum shear theory; the first governs until the shearing yield-point stress of the material is reached, after which the shear theory holds. The exact point of the change from one law to the other depends upon the ratio of the shearing yield-point stress to the yield-point stress in simple tension.

3. Stiffness.

Although elongations have been measured in the majority of tests heretofore made, no attempt seems to have been made to determine the law of stiffness. It has been taken for granted that the deductions of the mathematical theory of elasticity, as embodied in St. Venant's theory, held, or else no attention has been paid to the strains except as related to the strength of the material in the determination of the yield point or elastic limit. The weakness of the mathematical theory of elasticity lies in its generalization of Hooke's Law and the neglect of the temperature changes, so that the strains obtained with isotropic materials could but closely approximate the computed values. The effect of shear in producing strain has been neglected and is small under ordinary circumstances, but the variation of shear in different directions throughout the specimen, the possibility of a change in Poisson's ratio, the possibility of a different Poisson's ratio and modulus of elasticity with and across the direction of rolling or drawing, enter to complicate the problem.

From the general average curves of the axial and circumferential gage lines of tube number 5 it is easy to deduce the value of .334 for Poisson's ratio and 27,200,000 pounds per square inch for the modulus of elasticity. Lines have been drawn on the curves of general averages of the axial gage lines, (Figures

48 to 51) corresponding to the computed strains ε_{zz} and $\varepsilon_{\theta\theta}$, using the values obtained from tube number 5 for Poisson's ratio and the modulus of elasticity. It is seen that these lines agree quite closely with the observed values, except in the case of tubes number 1 and 7, the former showing a lower elongation and the latter a greater elongation than the computed values. Apparently within the range of application of the mathematical theory of elasticity--where E is constant--the elongations follow the theory with sufficient exactness to say that the theory holds. Lines have also been drawn to represent the strains corresponding to a stress condition in simple tension equal to the greater principal stress. The maximum shear theory does not concern itself with the accompanying strains.

Taking up the circumferential lines (pages 129 and 130) there have been drawn on the curves of general averages, lines representing the elongations accompanying an equivalent stress in simple tension and the mathematical theory of elasticity, with the values of Poisson's ratio and modulus of elasticity as before. There is a decided change in the slope of the various lines, but the lines of the mathematical theory of elasticity do not seem to fit well. A careful inspection will show that it is necessary to use a different value for both the modulus of elasticity and Poisson's ratio, the latter requiring the greater change. An increase in Poisson's ratio will increase the strains of tube number 1 and lower those of the other tubes. An increase of the modulus of elasticity will lower all the strains proportionally. There is a strong possibility that both Poisson's ratio and the modulus of elasticity vary in the two directions,

with and across the direction of the drawing. It is scarcely probable that the law changes, and the close agreement between the computed and observed values for the axial lines gives strong support to the belief that there the strains follow the mathematical theory of elasticity. However, since the deformations in some instances are quite small, the error of observation in those cases is relatively large, so that deductions from such tests should be made with care. The indications are that the modulus of elasticity and Poisson's ratio are different in different directions throughout the steel, just as Bauschinger has shown the shearing strength of steel to vary in different directions. Tests to determine these points would throw an interesting side light on the interpretation of the results noted above.

The stress-strain diagrams for the circumferential lines of all the tubes whose stress ratio is less than .92 break approximately at the unit stress corresponding to the break in the curve of the axial gage lines of the same tube. These breaks are the effects of the increased axial elongations and are to be accounted for by Poisson's ratio remaining approximately constant after the yield-point has been passed. This means that the tendency to reduce the diameter of the tube is greater than the increase in diameter produced by the increase of the water pressure. Consequently the curve reverses and the strains decrease, or at least do not increase.

A comparison of the circumferential and axial stress-strain diagrams of tube number 5, shows that Poisson's ratio

diminishes slightly; in the range covered by the diagrams this reduction amounts to about 6% or 8%.

A comparison of the circumferential stress-strain curves of the tubes having a tension ratio of .92 in the axial and circumferential directions, with those of the other tubes shows that the former curves break much more quickly after the yield point is reached, while the latter suffer practically no change, having nearly constant abscissas. The explanation of the rapid break in the circumferential diagrams of tubes number 6, 8, and 3--tension ratio..92--is that the shearing strains become an important factor. The slight decrease in Poisson's ratio would tend to increase the elongations, but the total change would amount to only two per cent and the real cause must be found elsewhere. The diagrams of tube number 4 show the same rapid break, while for the stress ratios less than .50 the circumferential diagrams break in the opposite direction. These things cannot be accounted for by Poisson's ratio alone, for this is practically constant, while the elongations of the axial lines of the tubes having a stress ratio of .50 or less(except tube Number 2) are increasing at an increasing rate and the circumferential elongations are at a stand-still. Under the conditions that exist below the yield point, the two diagrams--axial and circumferential--follow two straight lines of different slope, then when the yield-point stress is reached the elongations of the circumferential gage lines would follow a curve of constant ratio to the axial gage line elongations, if Poisson's ratio were the only factor. But this is not the case since the curves do not follow this proportionate curve, but instead they follow a straight line closely. The curves that continue straight

up and those that continue to increase can be divided sharply into those that are taken from the tubes whose yield-point stress lies on the horizontal shear line of the diagrams on page 131 and a second class which are taken from the tubes whose yield-point stress lies on the maximum strain theory line. This fact alone seems sufficient to account for the character of the curves by the action of shear in producing strain. Otherwise why should the curves of tubes number 2 and 4 turn in a different direction than those of tubes number 1 and 7?

For tube number 3, the stresses in the circumferential gage lines reached the yield point at the same unit stress as the axial gage line curves, but the applied loads were greater, since the ratio of tensions was .92. While the elongations along the axial lines increased more rapidly than those of the circumferential lines, just before the latter reached the yield point, the effect of shear up to this point must have been small because of the very gradual curvature of the axial stress-strain curves up to the stress corresponding to the applied load which produced yielding circumferentially. Also, the shear which causes yielding in an axial direction is on a different plane from that causing yielding in a circumferential direction. The former shear acts along a plane which passes through the line of the hoop tension and cuts the axis of the tube at an angle of 45 degrees. The latter shear acts on a plane which passes through the line of the axial tension and is parallel to the axis of the tube. These shearing stresses are of different magnitude according to the ratio of the stresses and each is equal to one half the principal stress cut by its plane at an angle of 45 degrees.

That the shearing stress accompanying the axial stress can affect the circumferential elongations is shown by the diagrams for the circumferential lines of tube number 4. It will be observed that in most cases the lines of the circumferential stress-strain diagrams continue straight for a short distance after the yield-point stress has been passed in the axial direction. This is during the stage intermediate between the elastic and plastic conditions. When the axial curves break sharply, the circumferential curves also break sharply. See the curves of tubes number 3 and 4, Figures 34, 35, 42, and 43. If Poisson's ratio were the only factor, all curves (with the possible exception of those of tubes number 3, 6, and 8) would curve to the left as the axial strains increased after the yield-point stress in the axial direction had been passed. But the curves of tubes number 2 and 4 turn toward the right, (they show increasing tube diameter) and since it is possible for the shearing strains, which are relatively large at this time, to have components, it seems that the shearing strains are important factors in producing circumferential elongations. Without such an effect, these curves would break in such a way that the rate of change of the elongations, at least, would decrease, and possibly the elongations would diminish, for the hoop tension is well below the yield-point stress.

It must be concluded that above the yield-point stress Poisson's ratio decreases slightly and the effect of shearing strains is noticeable. Further, the elongations follow the mathematical theory of elasticity for all stress ratios, especially for the elongations in an axial direction, when the values of Poisson's

ratio and the modulus of elasticity in that direction are used in the computation, but different values of these two quantities exist for the circumferential direction and, judging from the observed elongations, both are higher.

4. Correlation of the Results of Other Investigators.

Before attempting a correlation of the work of others with the theory herein advanced, attention will be directed to several points of difference between the method of investigation here recorded and the methods used by others.

The greatest difference lies in the use of a portable apparatus to measure strains, the strain gage, whereby a large number of measurements could be taken, both along the specimen and around it. No assumption of stress distribution in any one direction need be made, for the strain gage records the variations and the gage length can be varied to suit the needs. By taking twenty-eight readings for every load increment, a certain positiveness of result is attained which is impossible with attached instruments. Local effects are thus minimized since many gage lines are used. Another difference lies in the larger size of the specimen used and the ratio of thickness of tube wall to diameter--about .085 to 5.570. The stress was practically uniform over the entire surface of the specimen; there is no "helping" effect by understressed material, no point of maximum stress to be located, while the use of Johnson's apparent elastic limit gives a definite point for comparison.

An attempt was made to keep a definite ratio of stresses throughout the test of a tube, so that comparison could be made later according to these ratios. As far as possible, it was intended to cover the entire range within the tension-tension quadrant. The experiments reported by others and referred to in

this section show generally a haphazard ratio, and the load was such that a definite stress was produced in one direction and then the other stress increased until yielding took place. Thus, a torque of say 1000 pound inches would be applied and maintained while the tension or compression was increased until the yield-point stress was reached. This did not produce a constant ratio and any error in determining the yield-point stress affected the ratio as well.

The first important work was that reported by J. J. Guest* in 1900. The tests were made upon small steel, copper, and brass tubes about $1\frac{1}{4}$ inch outside diameter and varying in thickness from .025 inch to .034 inch. Tests were made in combined torsion and axial tension, torsion and hoop tension, and axial and hoop tension. No definite ratios of stress were used. The strains were measured by a two point extensometer and although it was attached to the outside of the tube, the full hydrostatic pressure was counted as a third principal stress. Other than the tests on tube number 1, there are but two tests where the stress ratio is .50 or less, and the tests on tube Number 1 and one of the others follows the maximum strain theory closely. Criticism is to be made of the repeated use of the same specimen, since the yield-point stress is raised by repeated loading beyond the yield-point stress of the first test. It is not stated whether the tubes were annealed between tests. The results are taken to justify the maximum shear theory and in the main they do, since the majority of the tests had a stress ratio between .50 and 1.00, within which limits all are agreed that the shear theory holds.

* Philosopical Magazine, 1900

The tests also show that the maximum shear developed is greater than one-half the yield-point stress in simple tension.

Following Guest comes the work of C.A.M.Smith,* W. A. Scoble,° E. L. Hancock,** and William Mason°° on bars in torsion and tension and on tubes.

All their results are used to justify the maximum shear theory, although with one exception--that of Scoble's tests reported in 1906--the maximum shear developed is greater than one-half the tension, which was also noted in Guest's tests. The majority of these tests--like Guest's--are in the region where the stress ratio is greater than .50. These tests cover the entire four quadrants of combined stress and the results of Smith's and Hancock's tests have been plotted in Figures 55 and 56 on pages 134 and 135. On account of the symmetry of the diagram for compression and tension, but one side has been used and the results have been changed proportionally to compare them with a single set of theoretical curves. A comparison of the dual law herein proposed with the experimental results of these investigators, shows that the results fit this law better than the maximum shear theory which they are supposed to prove. Figure 55 shows the results of C. A. M. Smith's tests on S. S. and A. D. steel. Professor Smith maintains that the shear yield-point of steel is one-half the tensile yield point within small limits and quotes Turner's# tests to prove his point. His own tests do not bear out this statement and Turner's tests show considerable variation, averaging

* Inst. Mech. Engrs. 1909
° Phil. Mag. 1906
** Am. Soc. Test. Mat. 1905-6-7-8
°° Inst. Mech Engr. 1909
\# Eng'g, London Feb. 5, 1909

about .54. Professor Smith's tests are examples of careful work, but the interpretation of an unqualified endorsement of the maximum shear law cannot be accepted.

Mr. Scoble's results may possibly be accounted for by the way he located the yield point. This was taken at the intersection of the straight line of the elastic state with a line approximating the stress-strain diagram during the plastic state. Since the shear curve breaks more quickly than a tension curve, it may be that the yield points were affected differently. Again, his method of measuring the bending moment by means of the deflection of the beam may be in error, for the law of deflection may change when the bar is twisted as well as bent.

The results of Professor Hancock's tests are shown on page 135 in Figure 56, and in addition to the curve of the maximum shear theory and the maximum strain theory Hancock's ellipse has been added. In the figure the tests on one kind of steel are marked with the same figure, and for the different kinds of steel tested, reference is made to the proceedings of the A. S. T. M. 1908. Hancock used the p limit as his criterion. He alone of these investigators realized the short-comings of the maximum shear theory and endeavored to fit an ellipse to the experimental results. This ellipse fits quite closely, but the fact remains that while it is a close approximation, it does not fit the results as closely as do the curves representing the dual law. His ellipse is purely empirical, while the combination of the maximum strain theory with the maximum shear theory recognizes the basic principles of stress in materials and has a foundation in the theory of the strength of materials. It is not empirical,

and Hancock, in common with all the others, did not recognize the control by the two laws.

Since torsion combined with compression or tension can be resolved into a case of tension combined with compression, Smith's and Hancock's tests fall in the fourth quadrant and show the applicability of the two laws there.

Mason's tests on tubes in compression and internal pressure show that the maximum shear developed is greater than that in simple compression. The average of all of his tests, in which a constant stress ratio of 1 to 1 was used, gives this maximum shearing stress as .60 of the compressive yield-point stress. As all the tests had the one stress ratio, it is not possible to deduce a law from them, but it locates a point which coincides with the requirements of the dual law. These tests also serve to show in a striking manner the higher value of the yield-point stress when obtained by the drop of the beam rather than by means of the extensometer. The time element is very important near the yield point and throughout the plastic state. If the load is applied too rapidly an artificial raising of the yield point is obtained, even though an extensometer is used.

W. J. Crawford,* in a series of tests of small flat plates, found that the maximum strain theory held, but gave only general results.

M. Malaval° reported that in experimenting on material for guns, he found that the material obeyed the maximum strain theory. His apparatus consisted of a tension specimen held in

* Proc. Royal Society of Edinburgh, 1911-12
° Inter. Assn. Test. Mat., 1912

the ordinary way and subjected to compression at right angles by two plates drawn together with bolts. Between the plates was a third plate, a large steel ball, and a device to measure the load.

The difficulty of experimental work demands that correlation be done carefully, and minor inconsistencies are to be expected, both on account of the variation in the material tested and on account of the apparatus used. The number of tests made by these experimenters is insufficient to completely establish any theory, but a careful study of the published data will lead to the conclusion that the control by the two laws, the theory here advanced, conforms more closely to the experimental results than any single law.

Minor criticisms might be made of the methods and conclusions of the investigators, but this is not deemed to be the chief province of this discussion, nor is it necessary. Rather should credit be given for the work done in blazing the way, and attention be called to the verification of the theory here advanced by the results of this investigation and those of other investigators. By reason of this verification it is thought that the dual law is more nearly correct than the laws heretofore accepted.

5. Summary and Conclusions.

The following summary is divided into two parts, one dealing with the method of the investigation, and the other with the deductions which have been made from the data. Inasmuch as this is the first investigation of combined stress wherein a portable strain measuring instrument, such as the straingage, has been used, considerable emphasis has been laid upon this fact, as well as upon the size of the specimens, which are much larger than any heretofore used. These conditions tend to give more trustworthy results.

The experimental conclusions are:

(1) That the use of a portable strain measuring instrument is a decided advantage, since it makes it possible to take a large number of readings for each increment of load, on different gage lines, obviating to a large extent local variations in the test piece.

(2) That the use of large tubes with thin walls gives quite uniform stress distribution, the yield point is more positively determined, and the effect of eccentricity of loading is less than with solid bars because of the larger diameter of the tube.

(3) That the thickness of the tube walls can be accurately determined.

(4) That flat plates in cross bending give uneven distribution of stress and are not satisfactory for bi-axial loading tests.

(6) That the results of the tests reported by previous investigators conform better to the dual law of strength than to any single law.

Fig. 23.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE D

SERIES 2

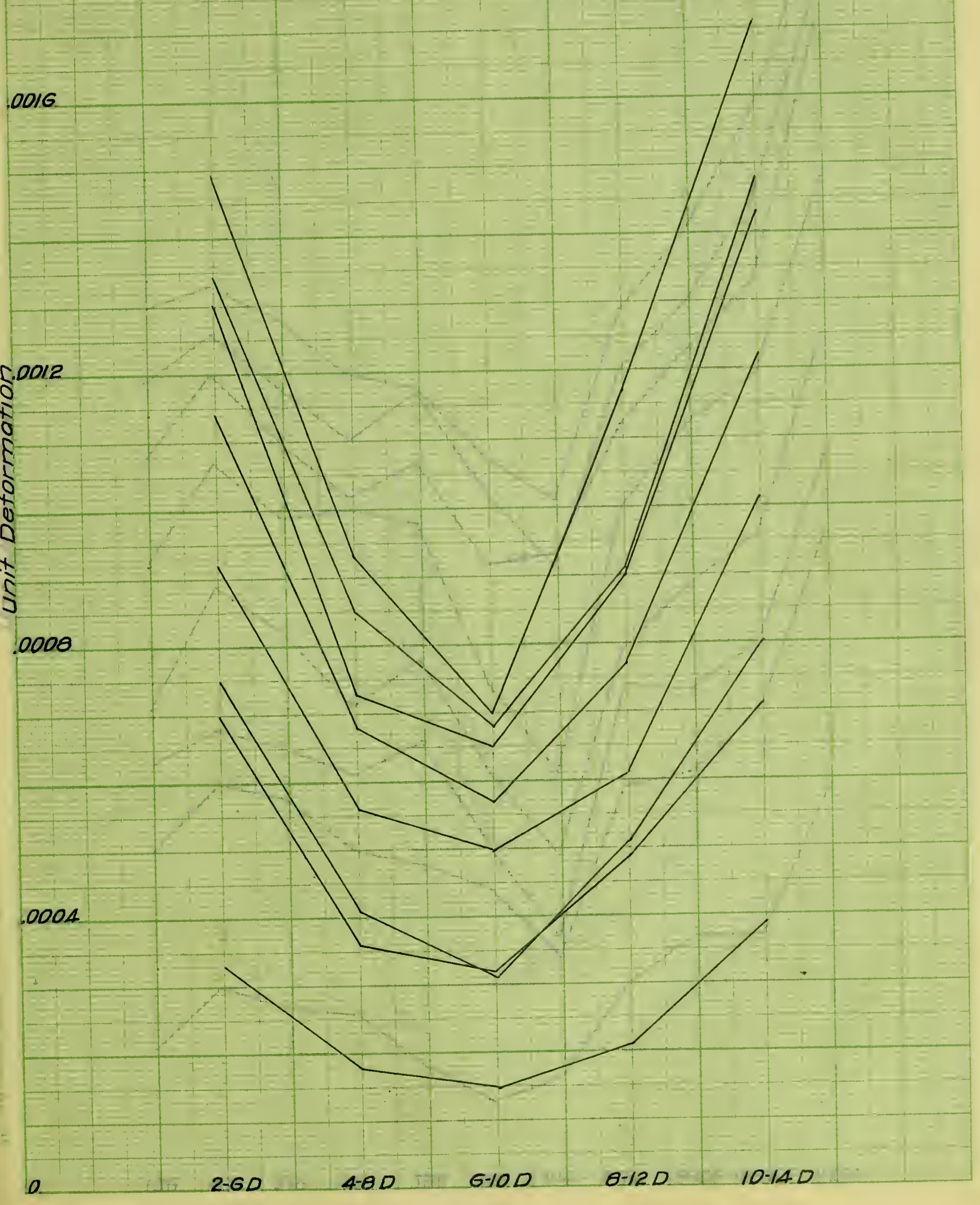


Fig. 23.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE D

SERIES 2

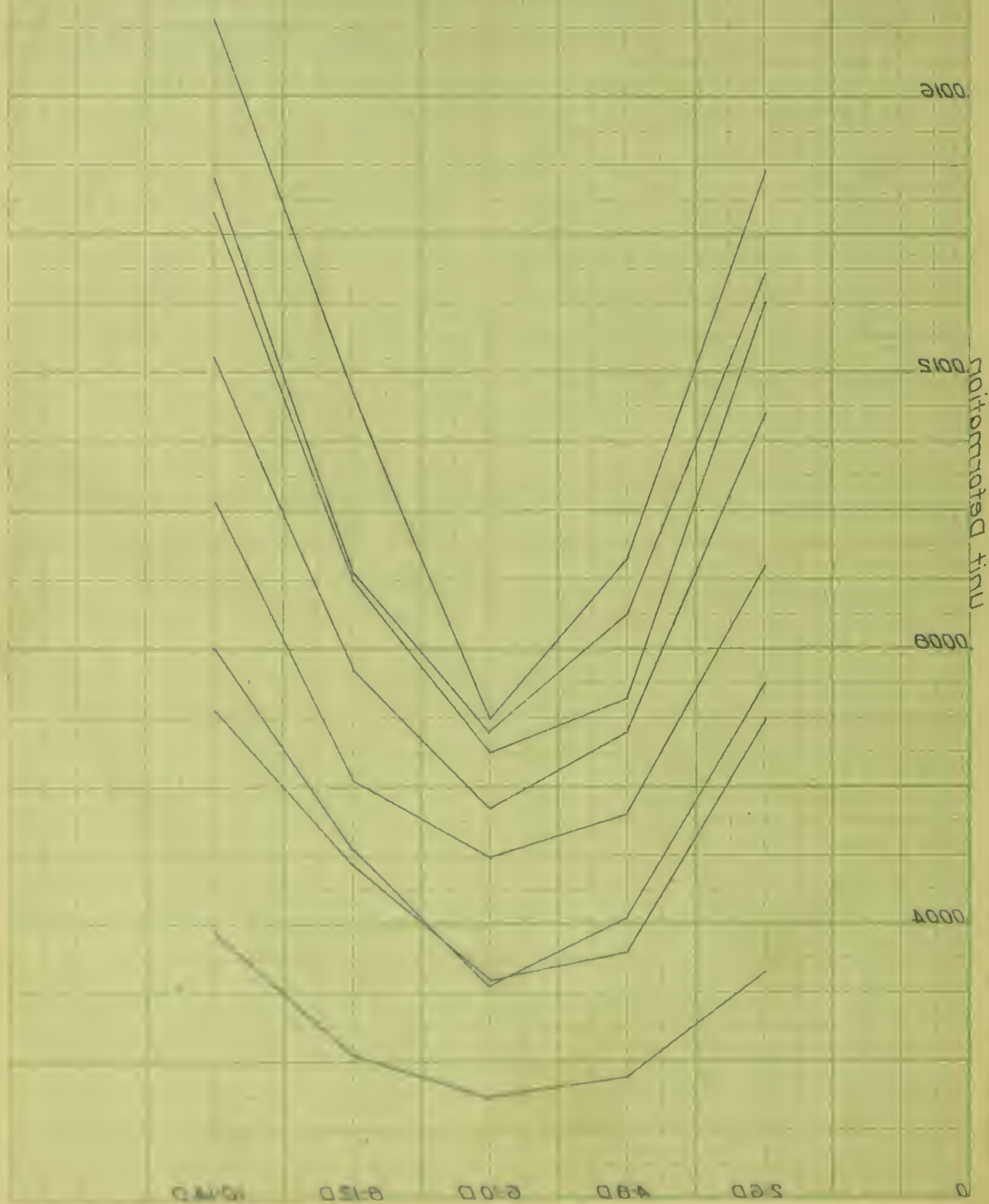



Fig. 24.
CROSS BENDING TESTS
DEFORMATIONS ALONG LINE G
SERIES 2
Unit Deformation
.0016
.0012
.0008
.0004
0
1-5G
2-6G
3-7G
4-8G
5-9G
6-10G
7-11G
8-12G
9-13G
10-14G
11-15G

Fig. 24.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE G

SERIES 2

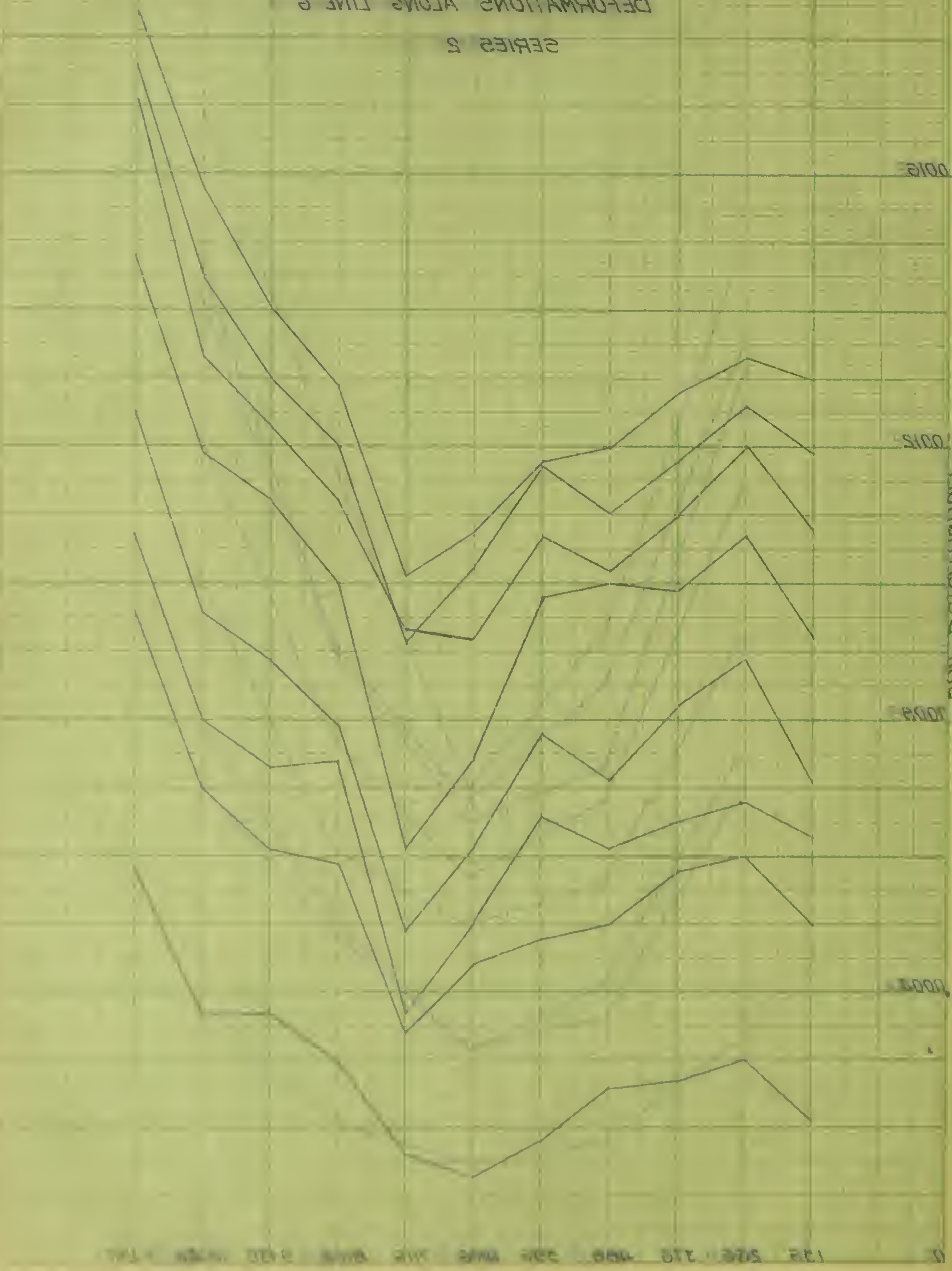
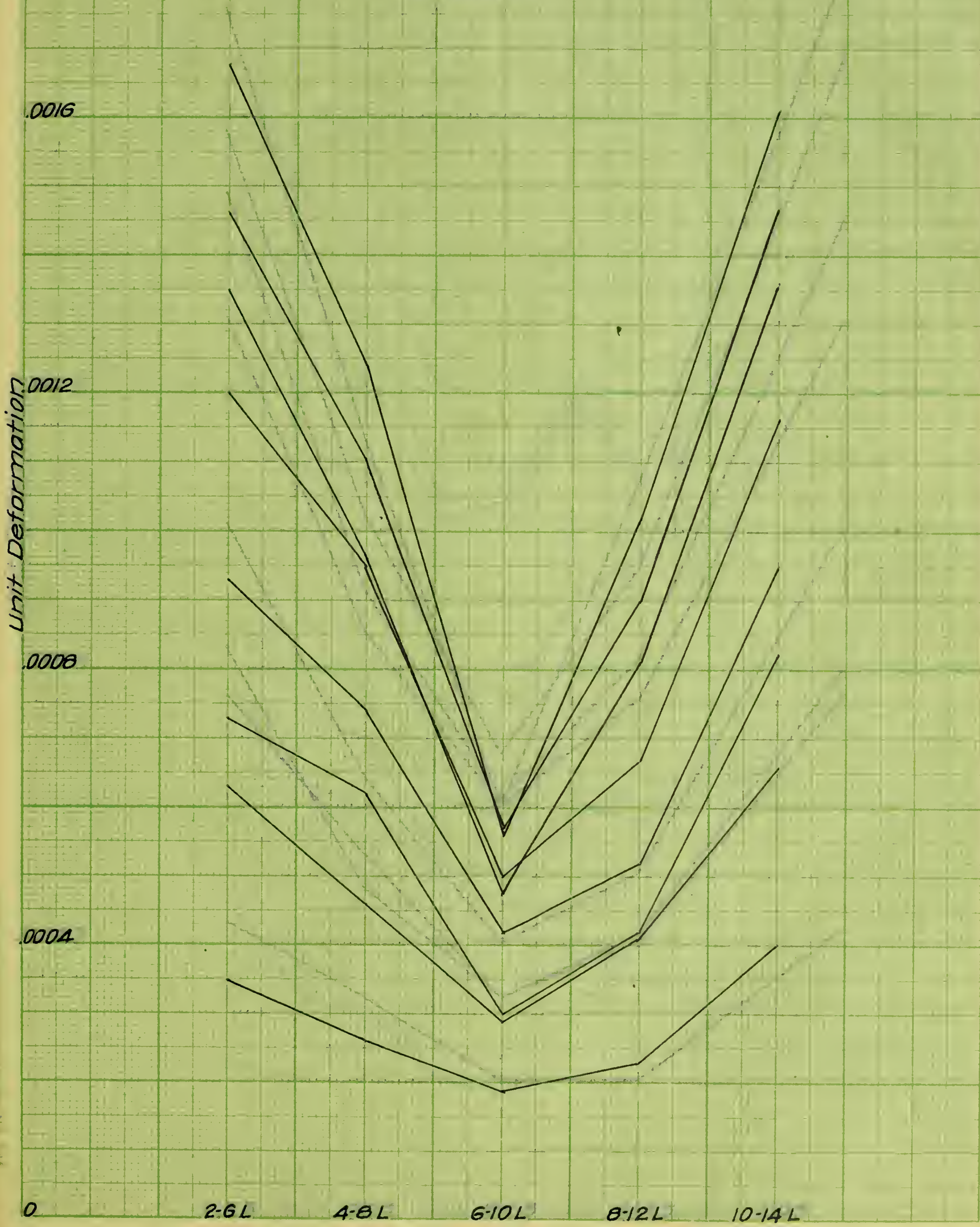


Fig. 25.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE L

SERIES 2



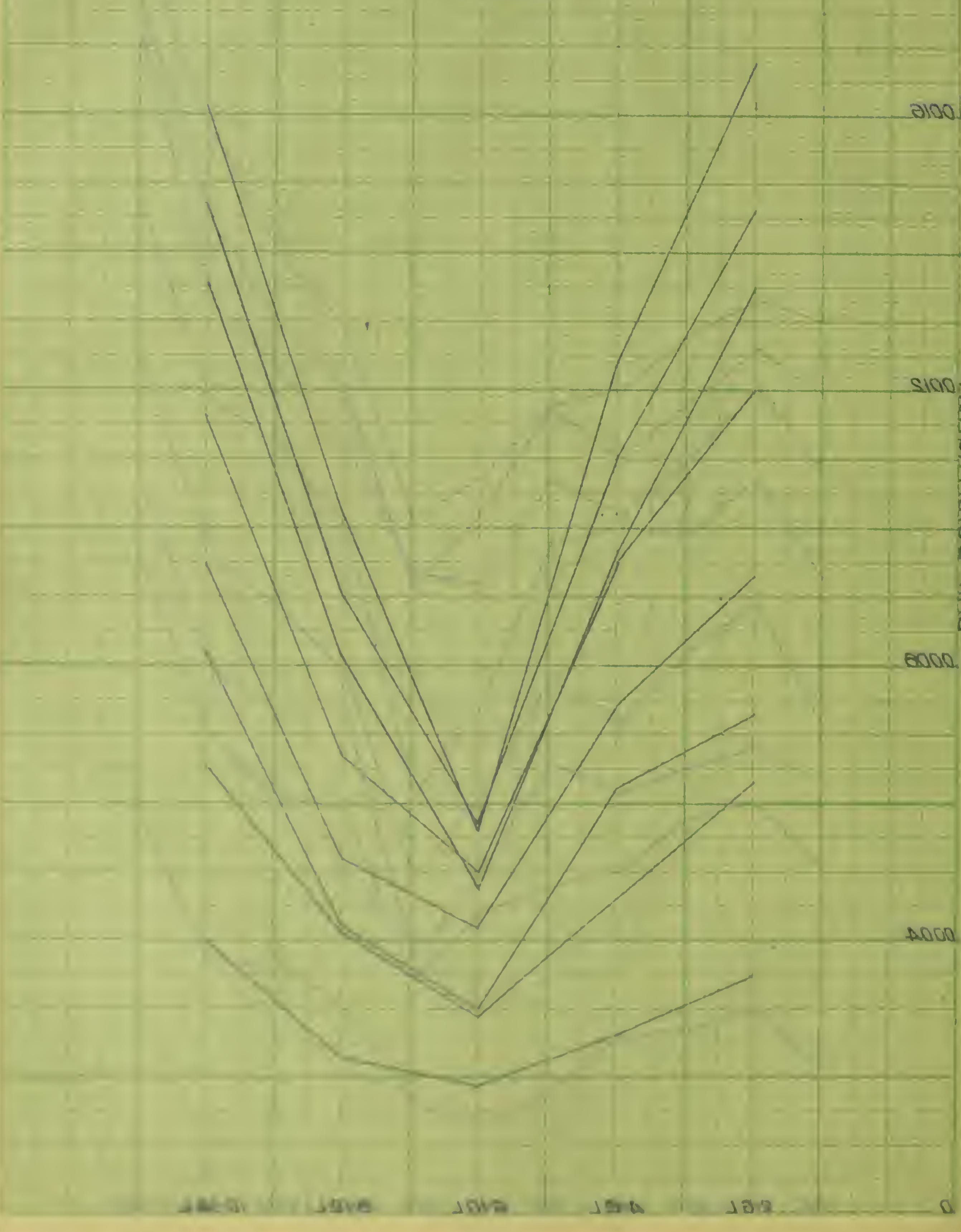
104

Fig. 25.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE L

SERIES 2



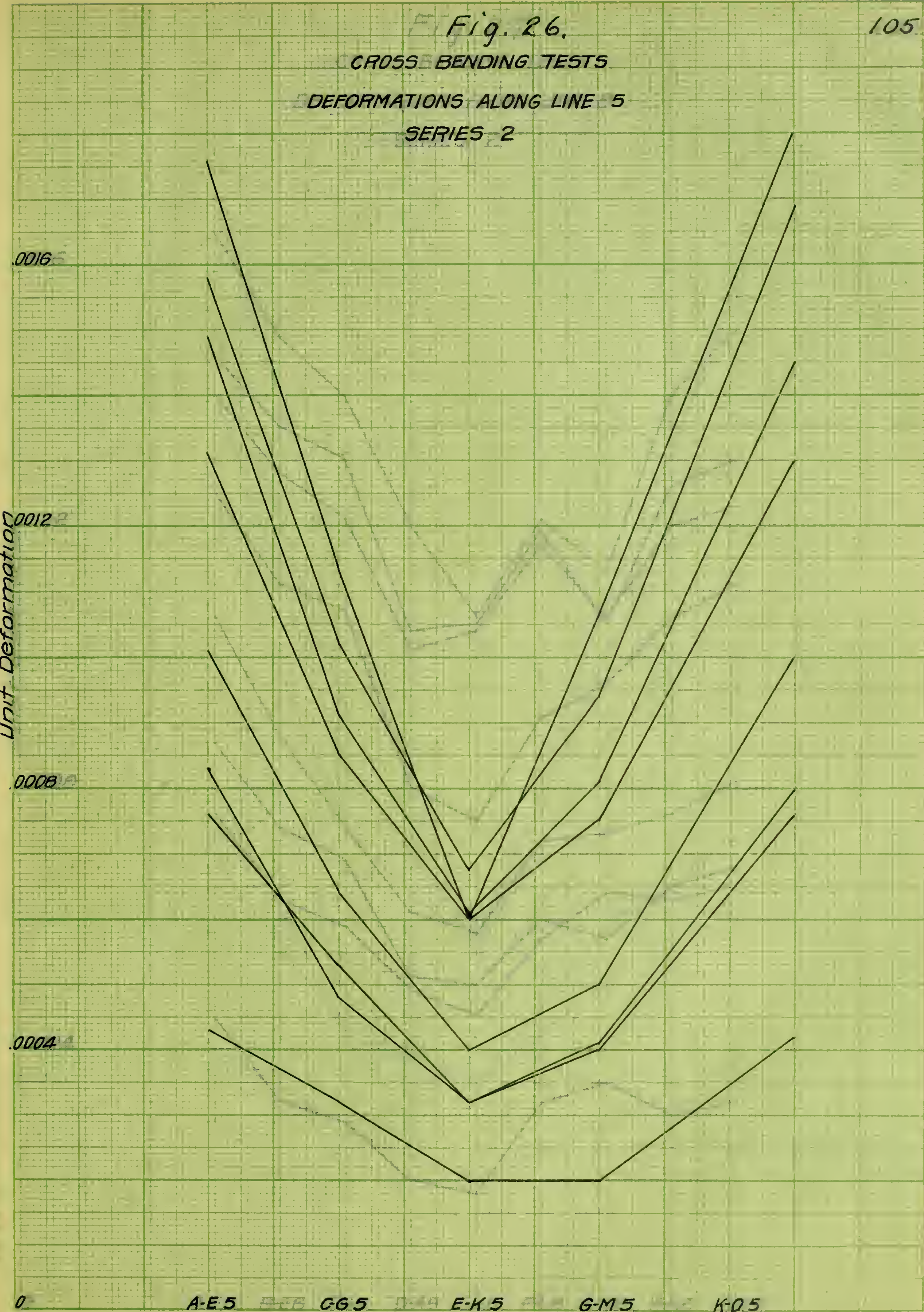


Fig. 26.
CROSS BENDING TESTS
DEFORMATIONS ALONG LINE 5
SERIES 2
Unit Deformation
.0016
.0012
.0008
.0004
0
A-E 5
C-G 5
E-K 5
G-M 5
K-O 5

Fig. 26.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE 5

SERIES 2

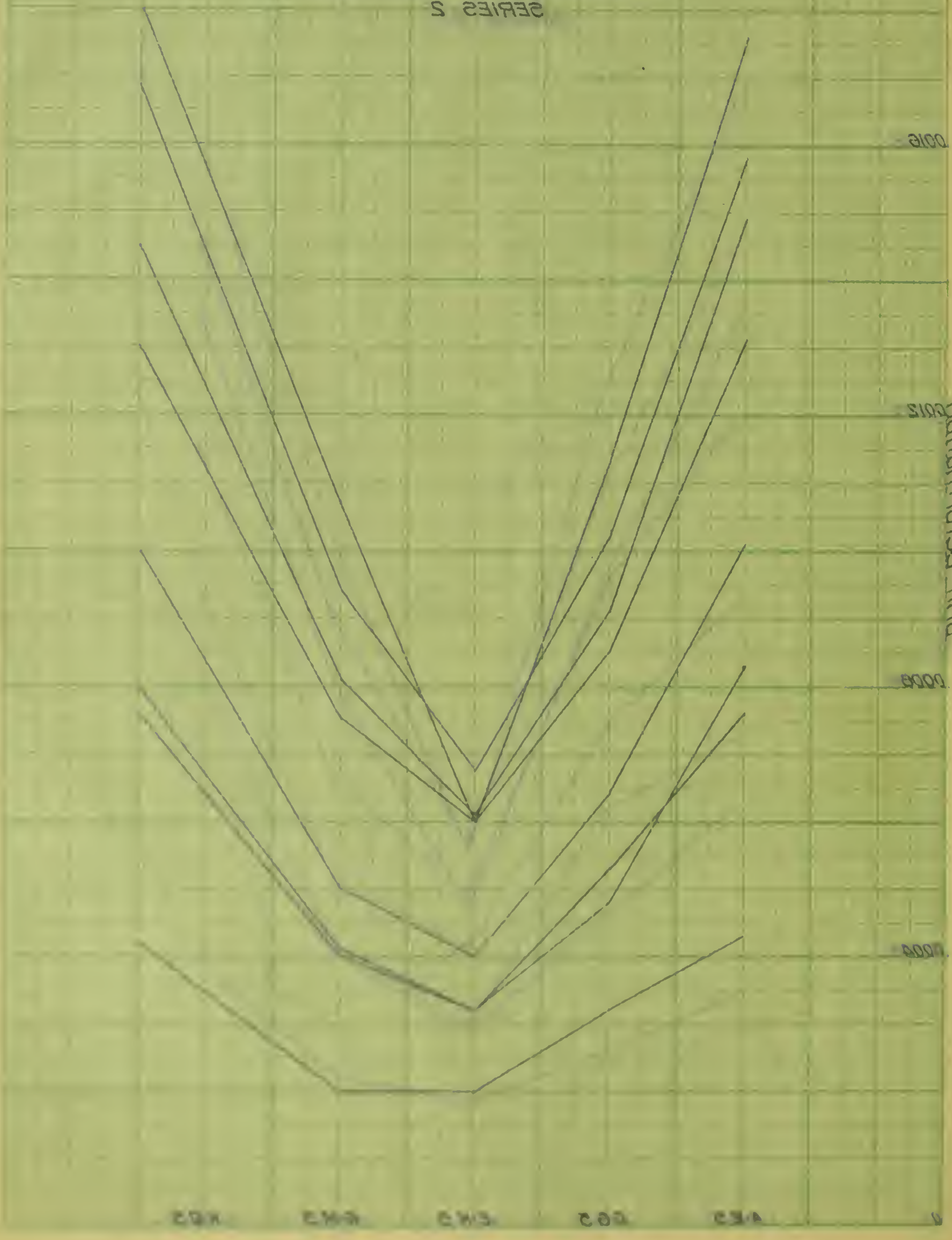


Fig. 27.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE 8

SERIES 2

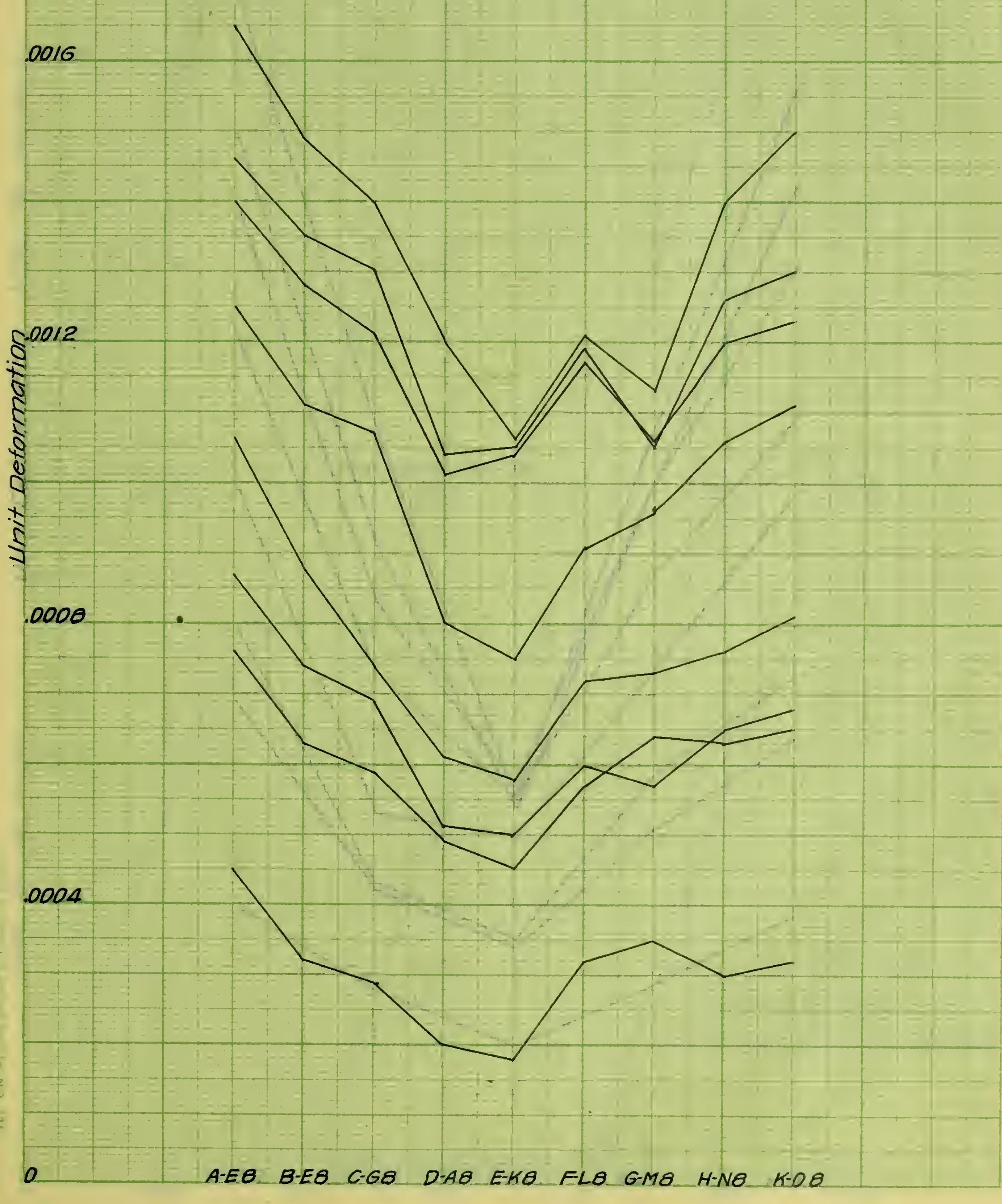


Fig 27.


CROSS BENDING TESTS
DEFORMATIONS ALONG LINE B
SERIES 2
Unit Deformation
.0016
.0012
.0008
.0004
0

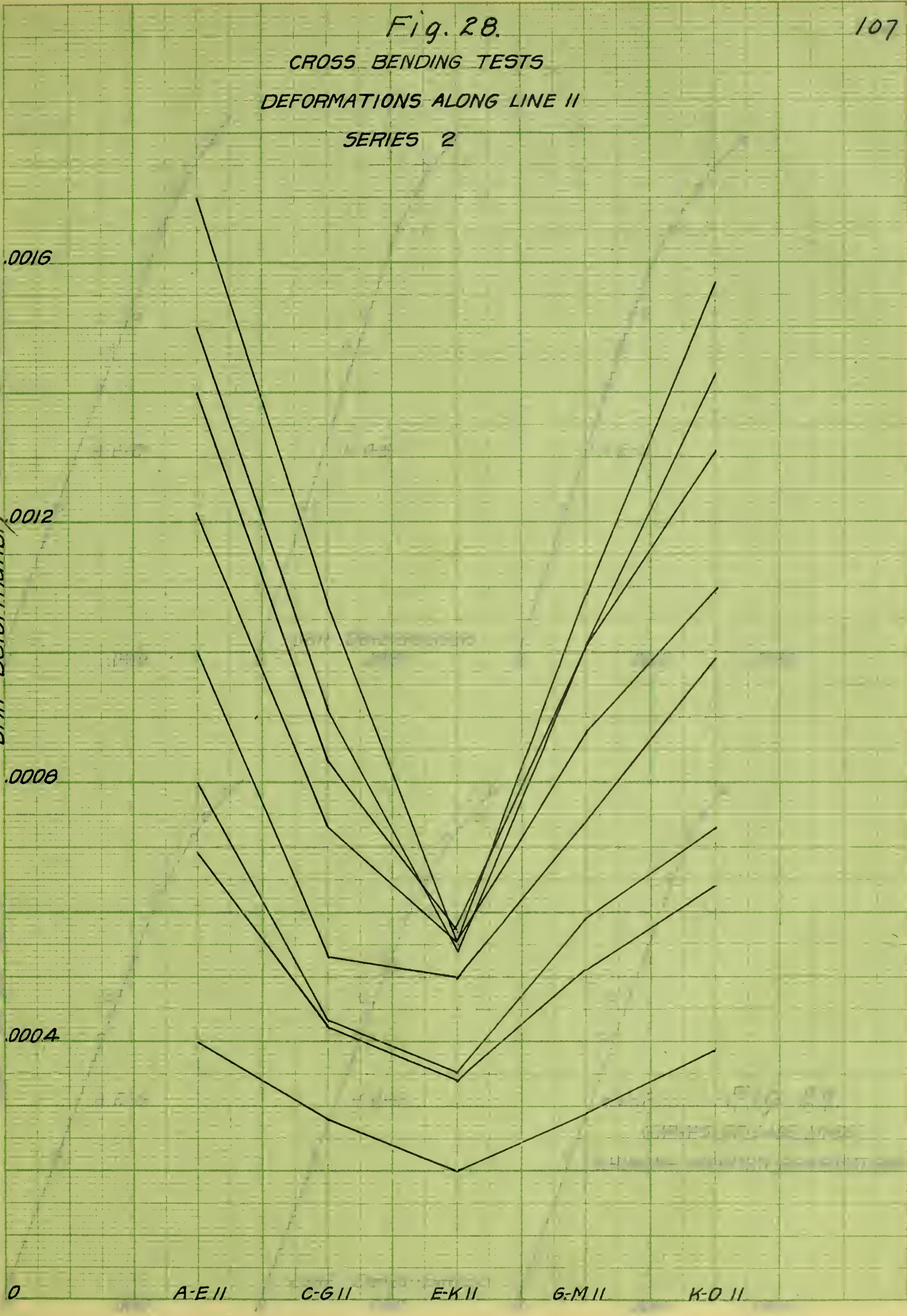


Fig. 28.
CROSS BENDING TESTS
DEFORMATIONS ALONG LINE II
SERIES 2
Unit Deformation
.0016
.0012
.0008
.0004
0
A-E II
C-G II
E-K II
G-M II
K-O II

Fig. 28.

CROSS BENDING TESTS

DEFORMATIONS ALONG LINE II

SERIES 2

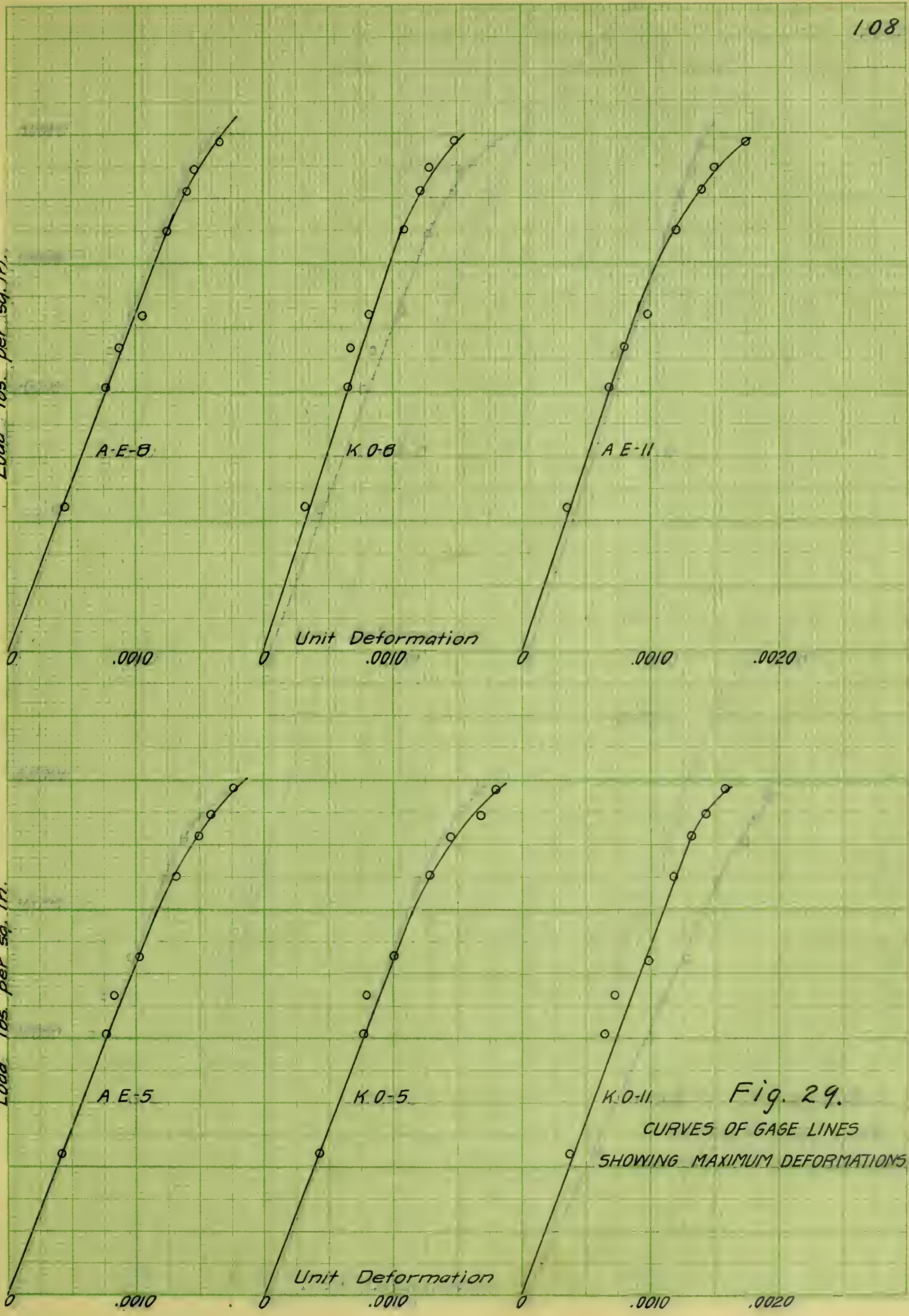


Fig. 29.
CURVES OF GAGE LINES
SHOWING MAXIMUM DEFORMATIONS

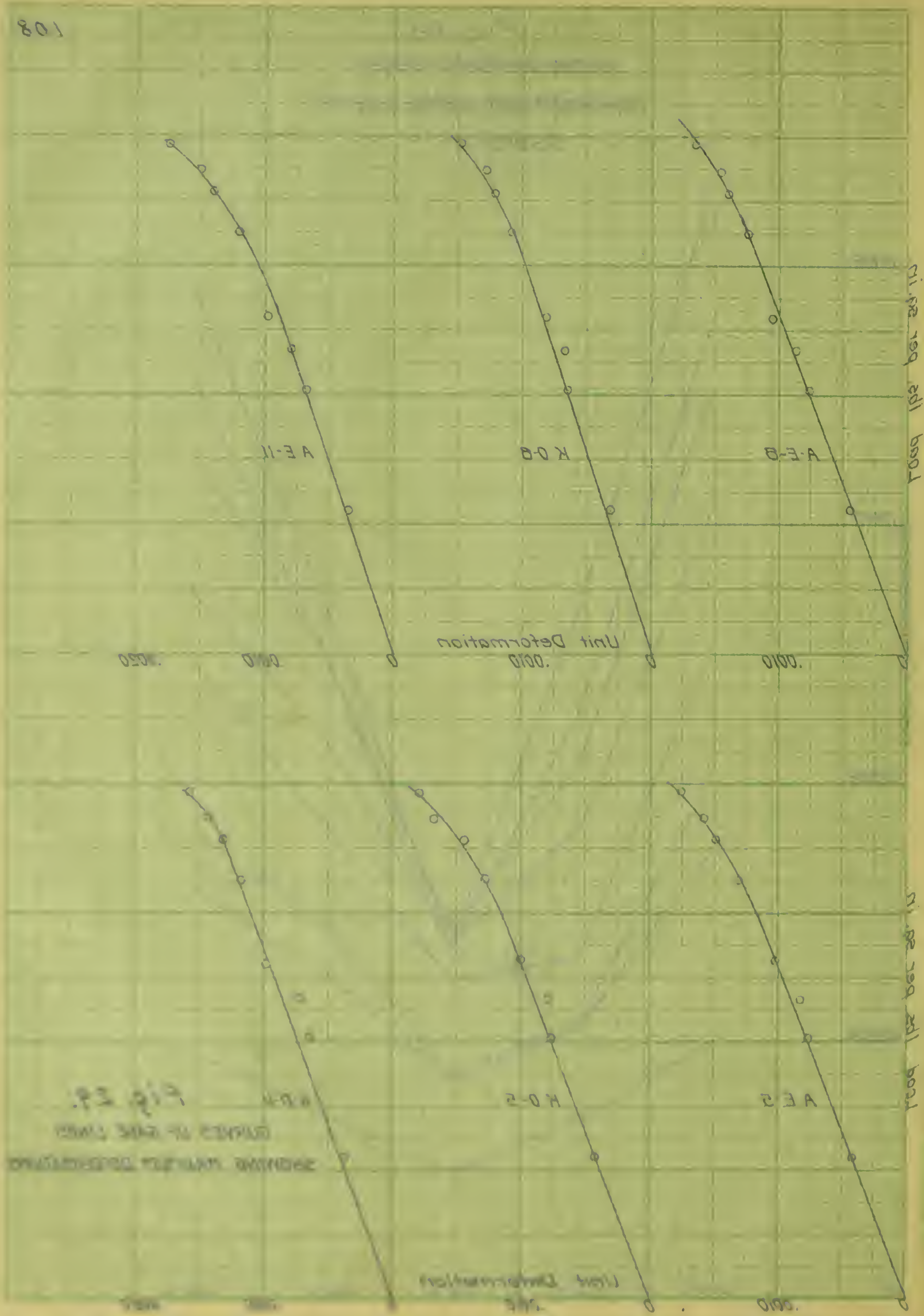

A E-11
K O-8
A-E-8
Unit Deformation
.0020
.0010
0
Load lbs per sq in
A E 5
K O-5
Unit Deformation

Fig. 29.

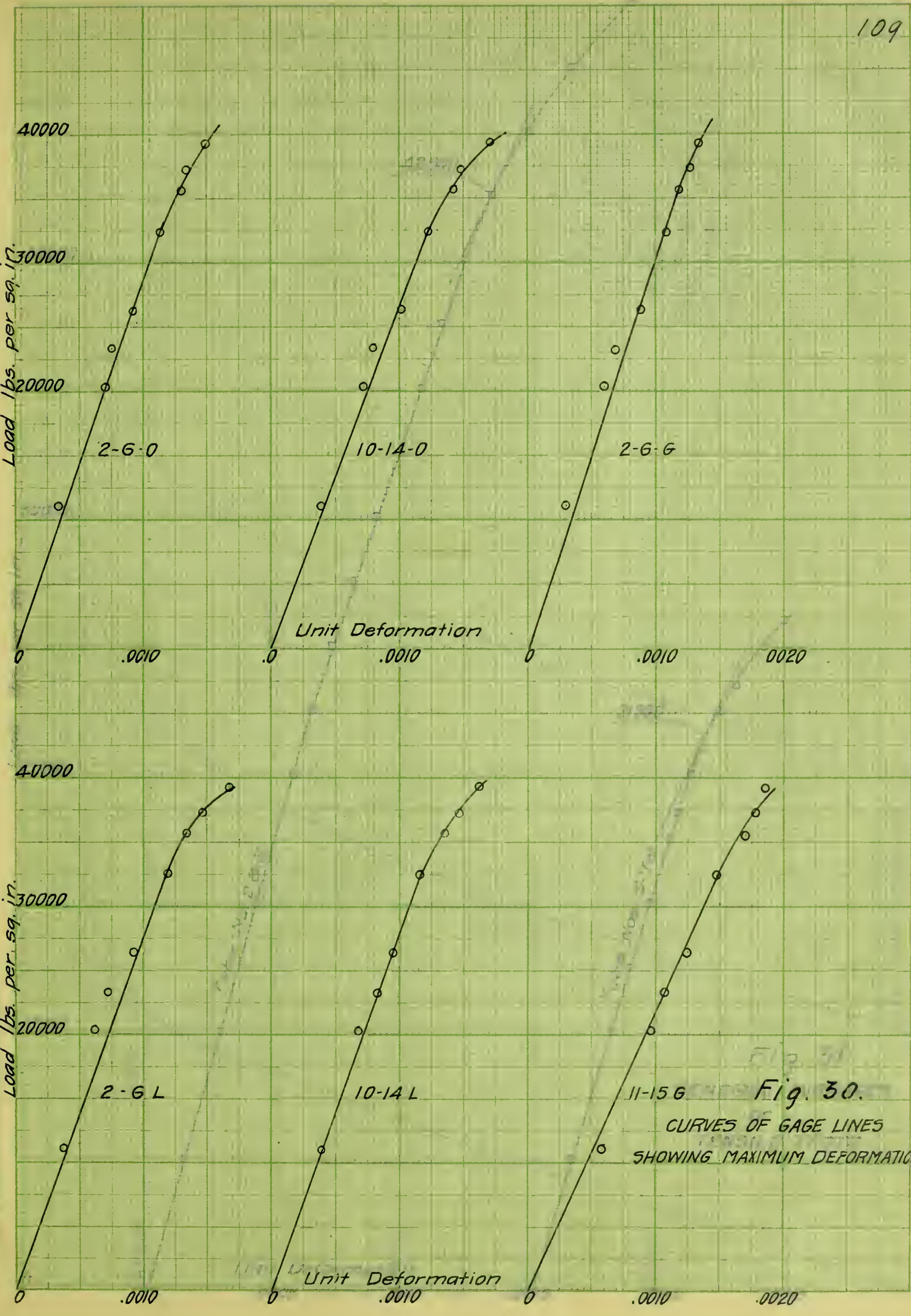
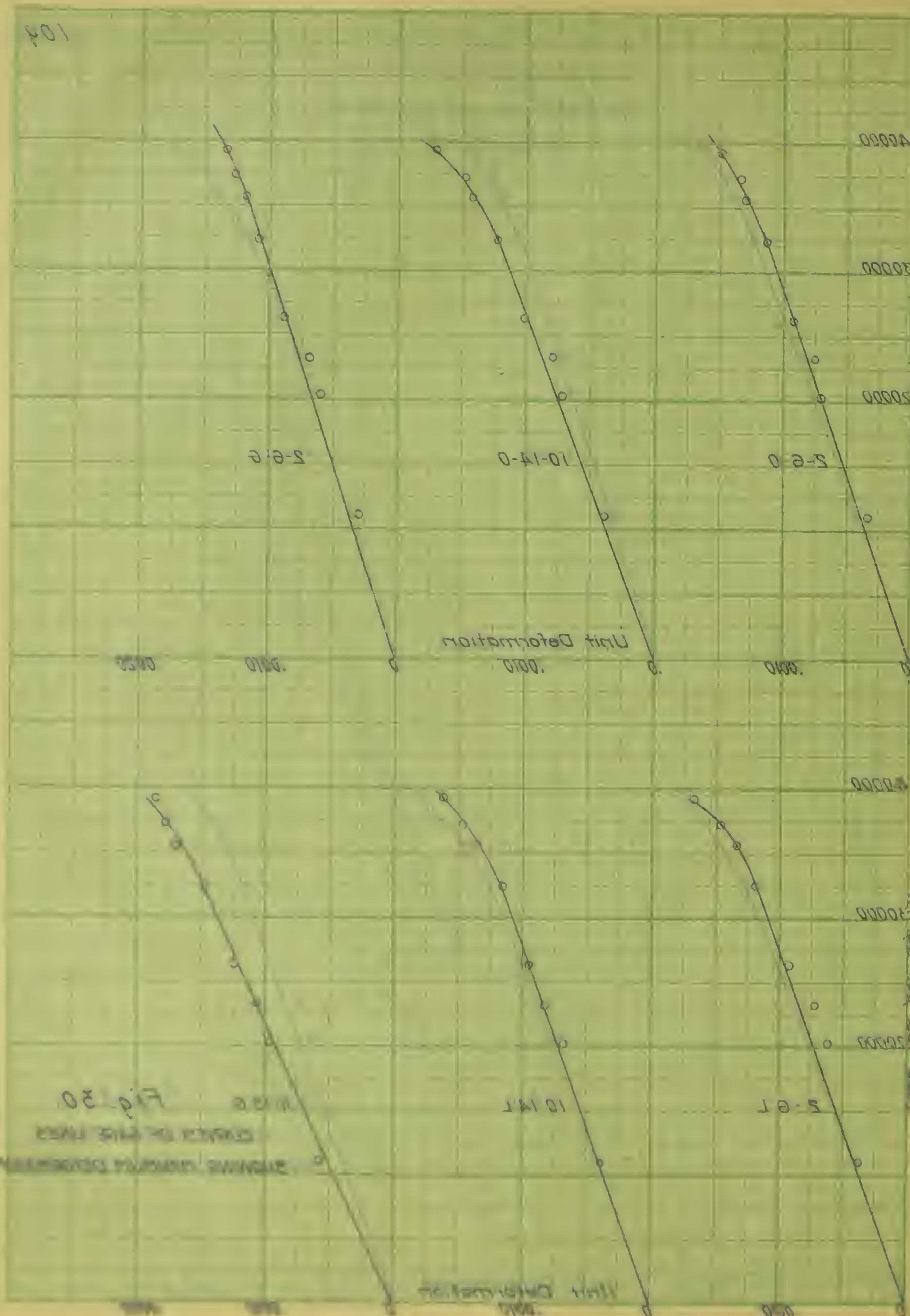


Fig. 30.
CURVES OF GAGE LINES
SHOWING MAXIMUM DEFORMATIONS


Load lbs. per sq. in.
40000
30000
20000
Unit Deformation
.0010
0
2-6-G
10-14-0
2-6-0
2-6-L
10 14 L
Fig. 30

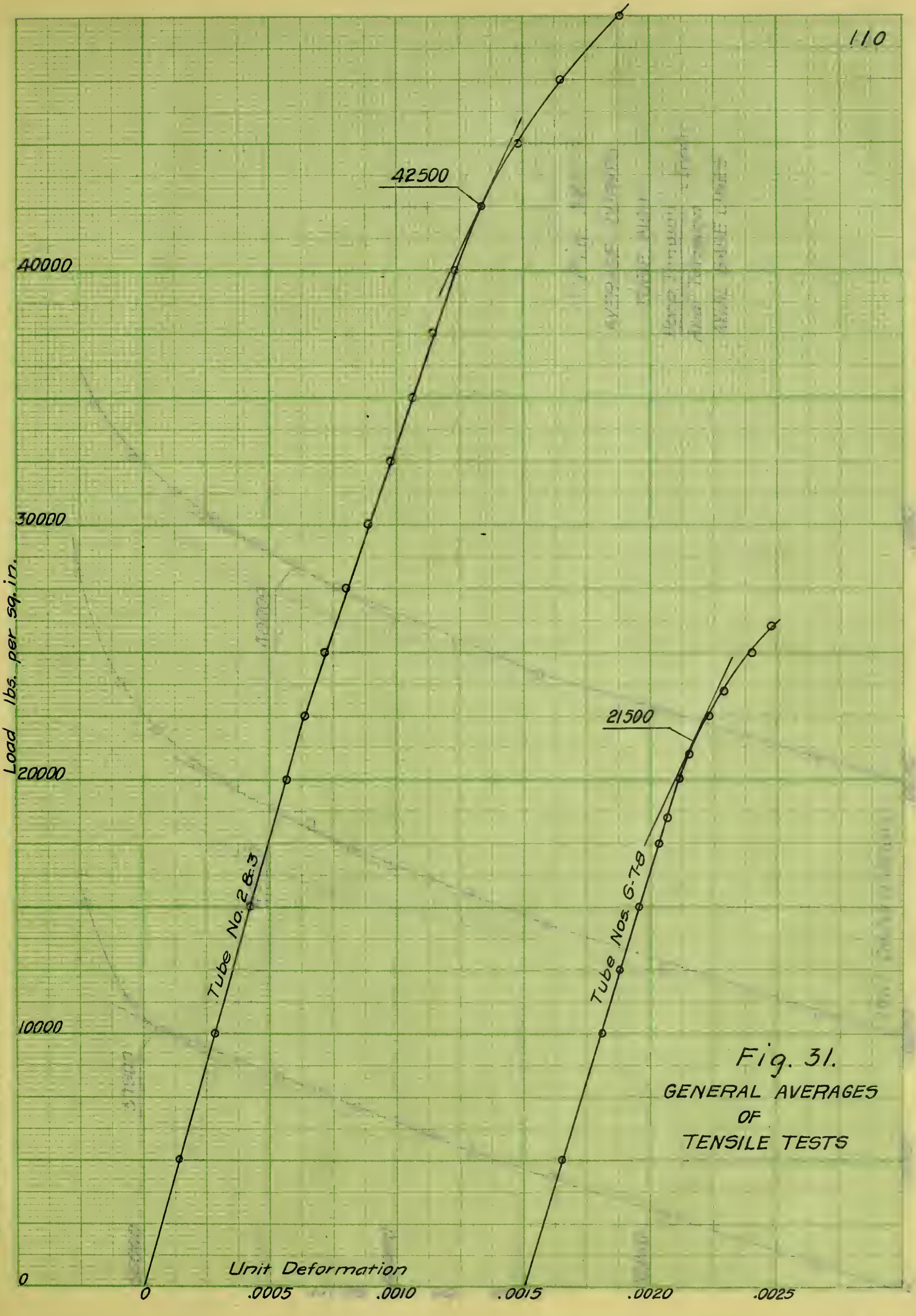
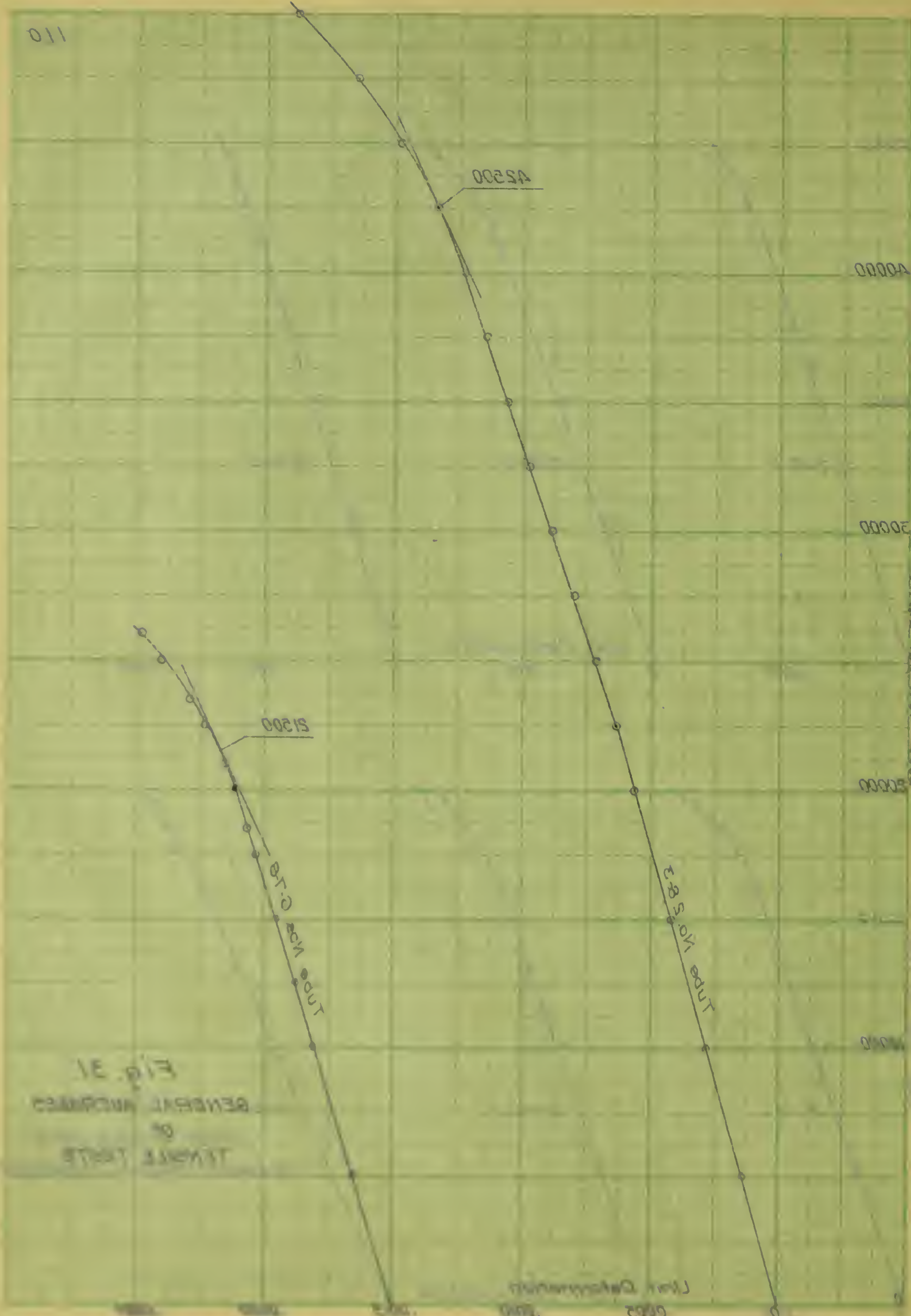
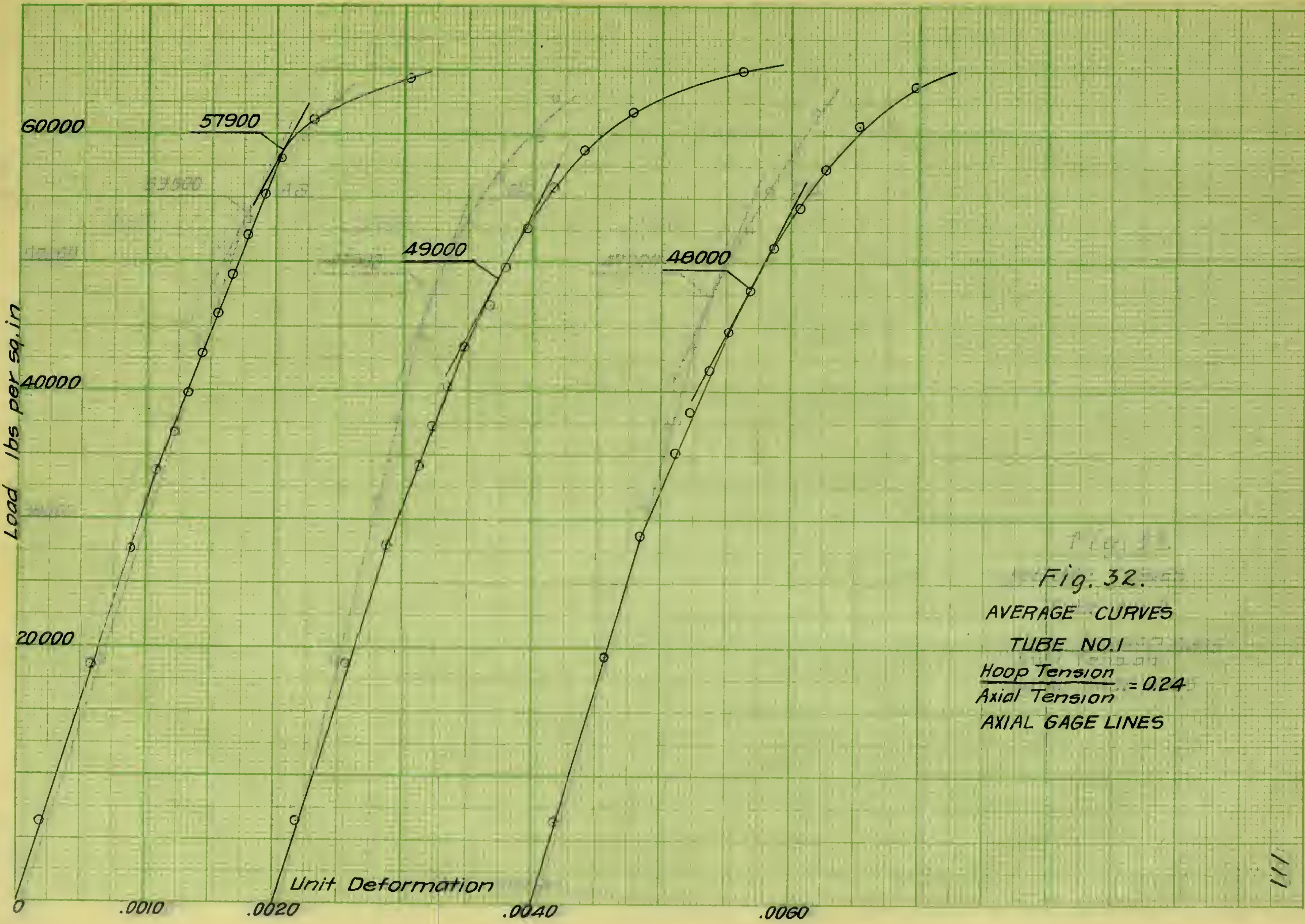


Fig. 31.
GENERAL AVERAGES
OF
TENSILE TESTS


42500
40000
30000
21500
20000
Fig. 31


57900
49000
48000
60000
40000
20000
Load lbs per sq. in
0
.0010
.0020
.0040
.0060
Unit Deformation
Fig. 32.
AVERAGE CURVES
TUBE NO. 1
Hoop Tension / Axial Tension = 0.24
AXIAL GAGE LINES

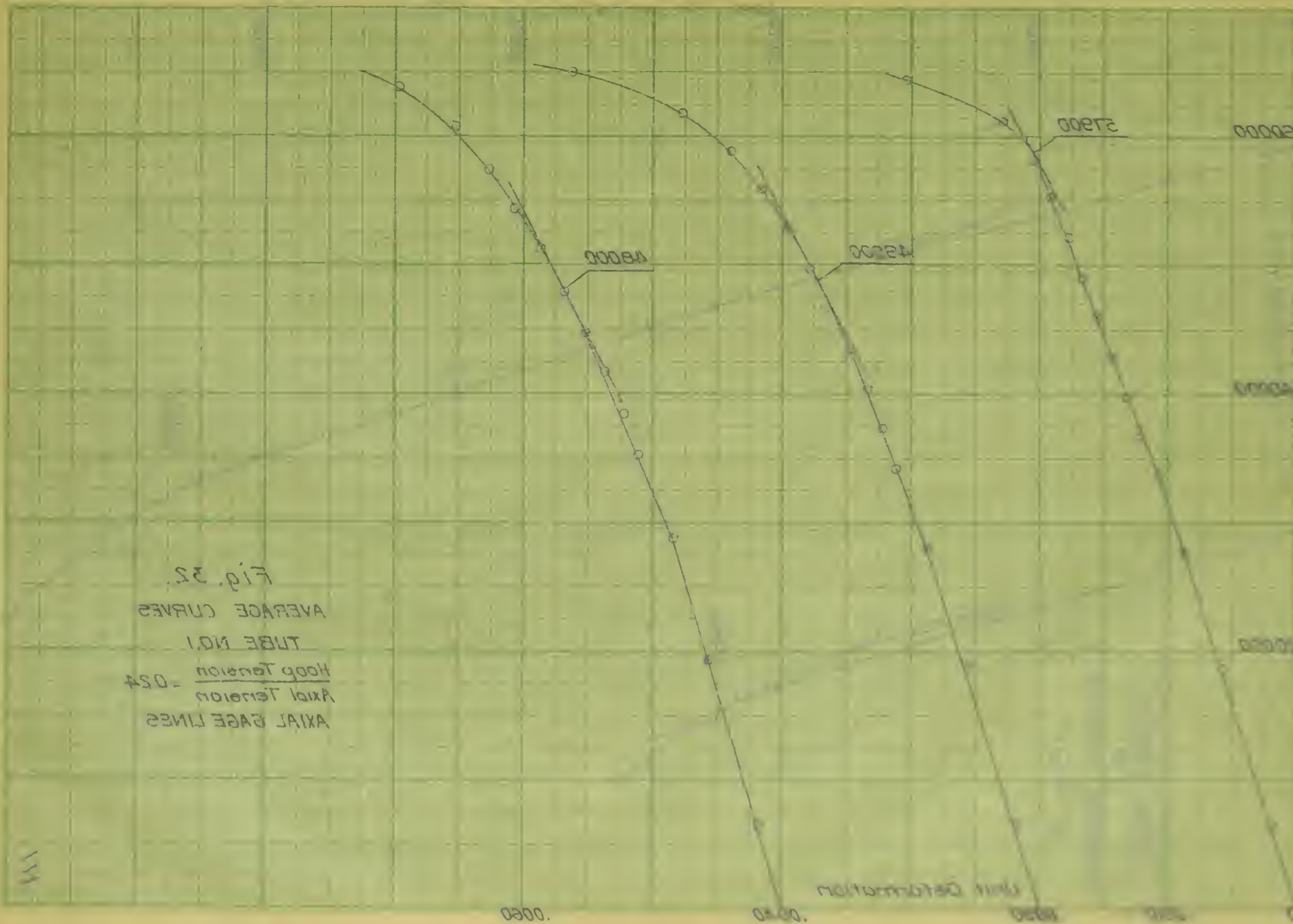


Fig. 52.
AVERAGE CURVES
TUBE NO.1
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = .024$
AXIAL GAGE LINES

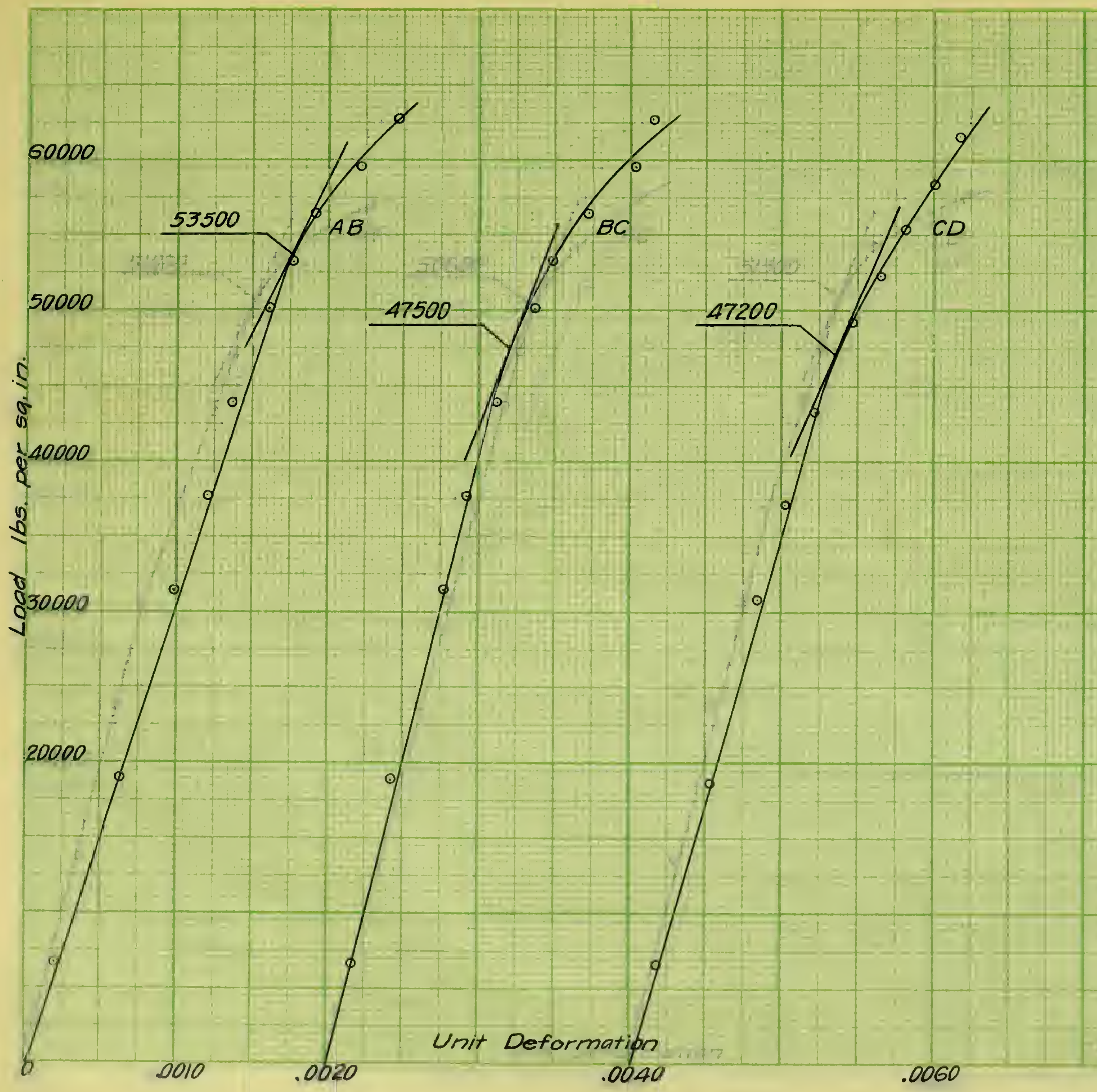


Fig. 33.
AVERAGE CURVES
TUBE NO. 2
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.475$
AXIAL GAGE LINES

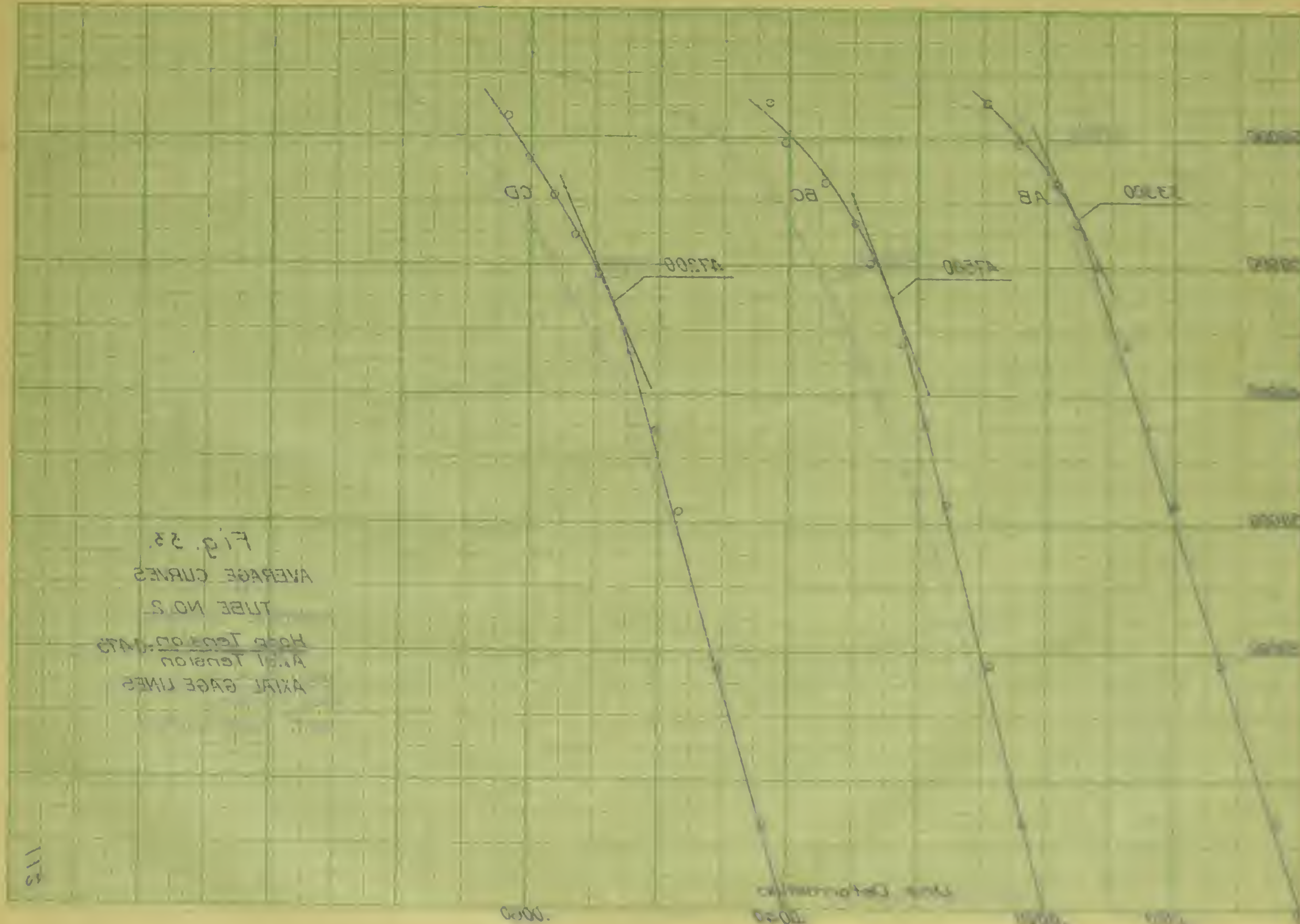


Fig. 35.
AVERAGE CURVES
TUBE NO. 2
AXIAL GAGE LINES
AB
BC
CD
13300
47540
47200

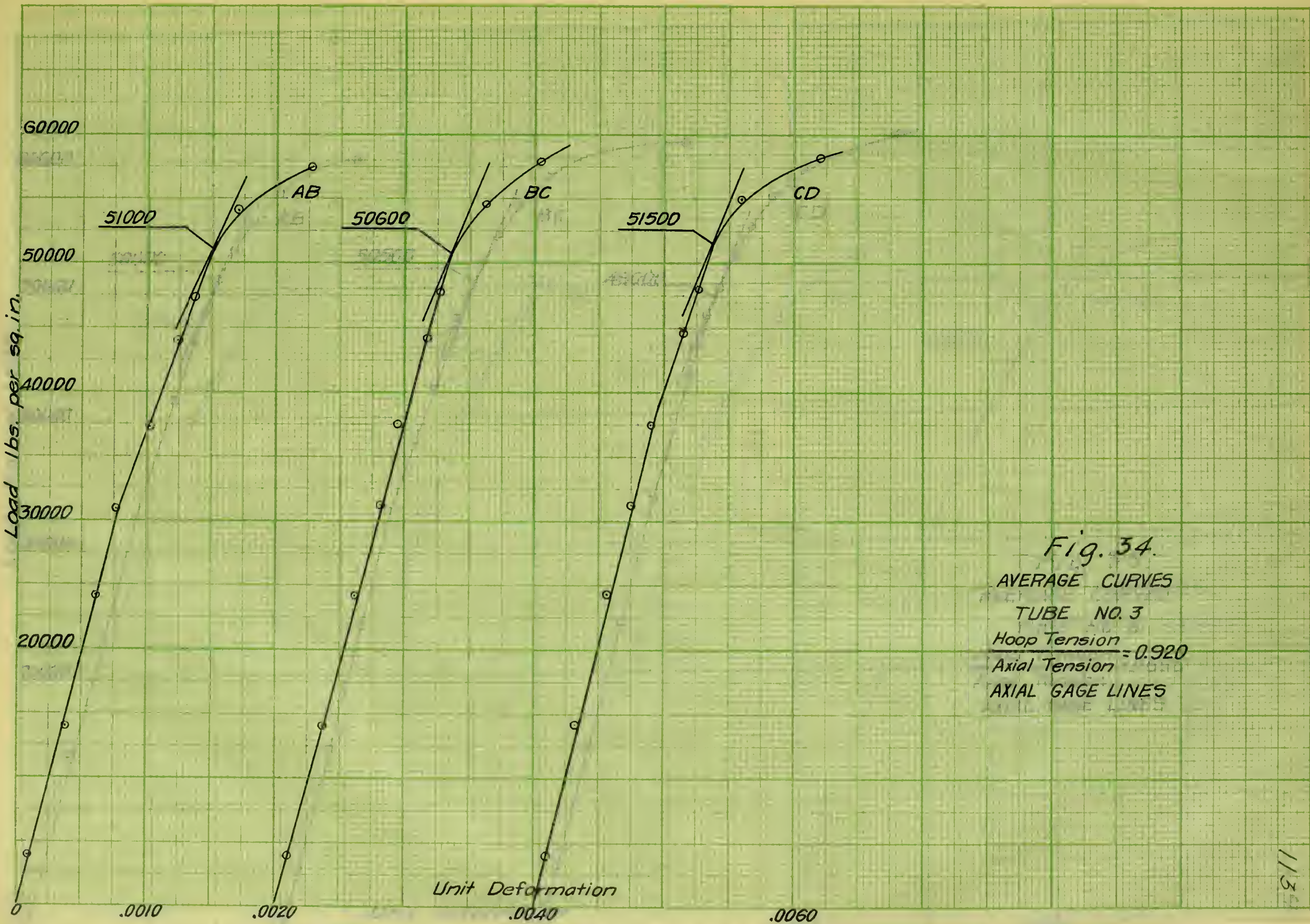


Fig. 34.
AVERAGE CURVES
TUBE NO. 3
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.920$
AXIAL GAGE LINES

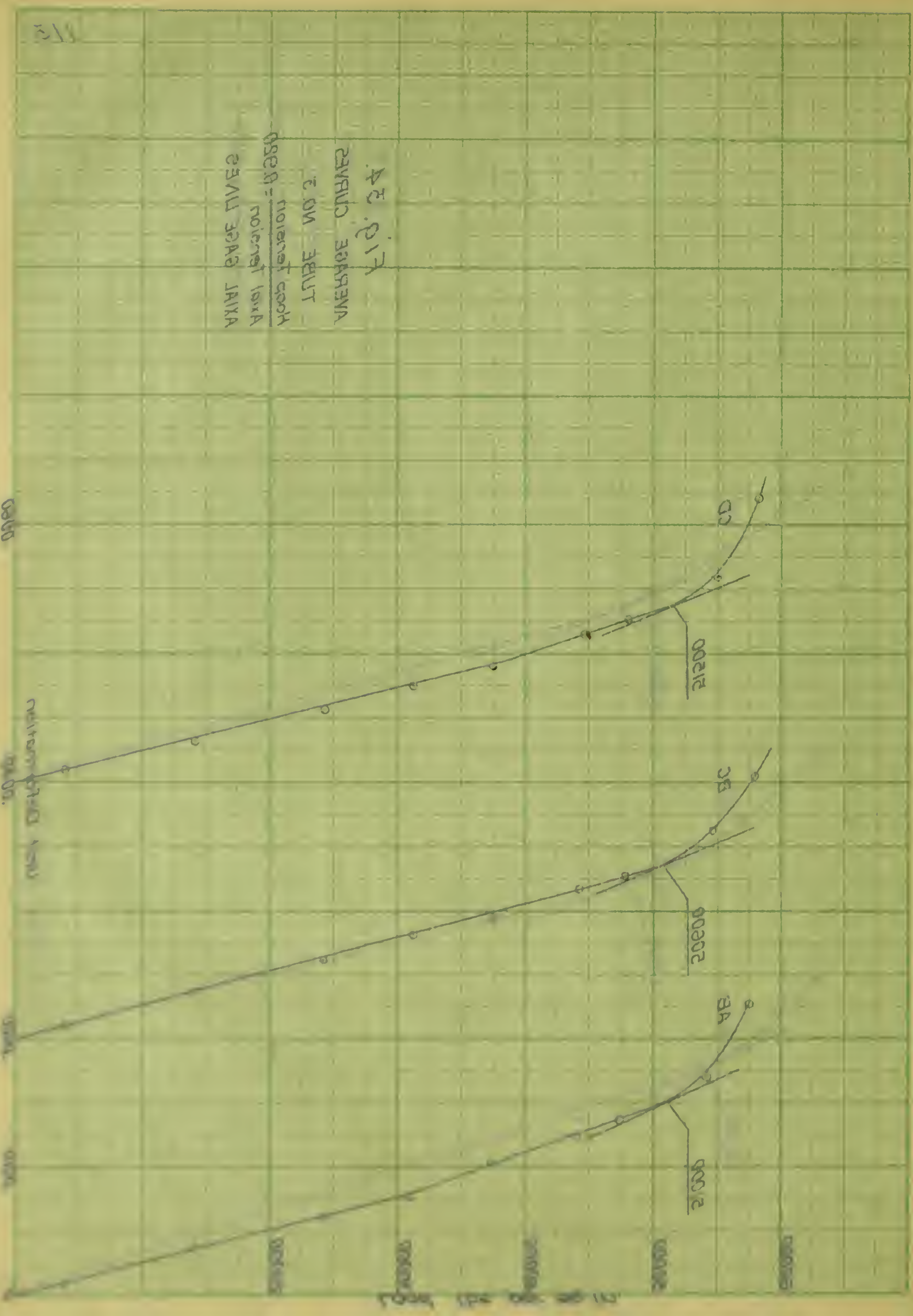


Fig. 34
AVERAGE CURVES
TUBE NO 3
AXIAL GAGE LINES
AB
BC
CD
50600
51500

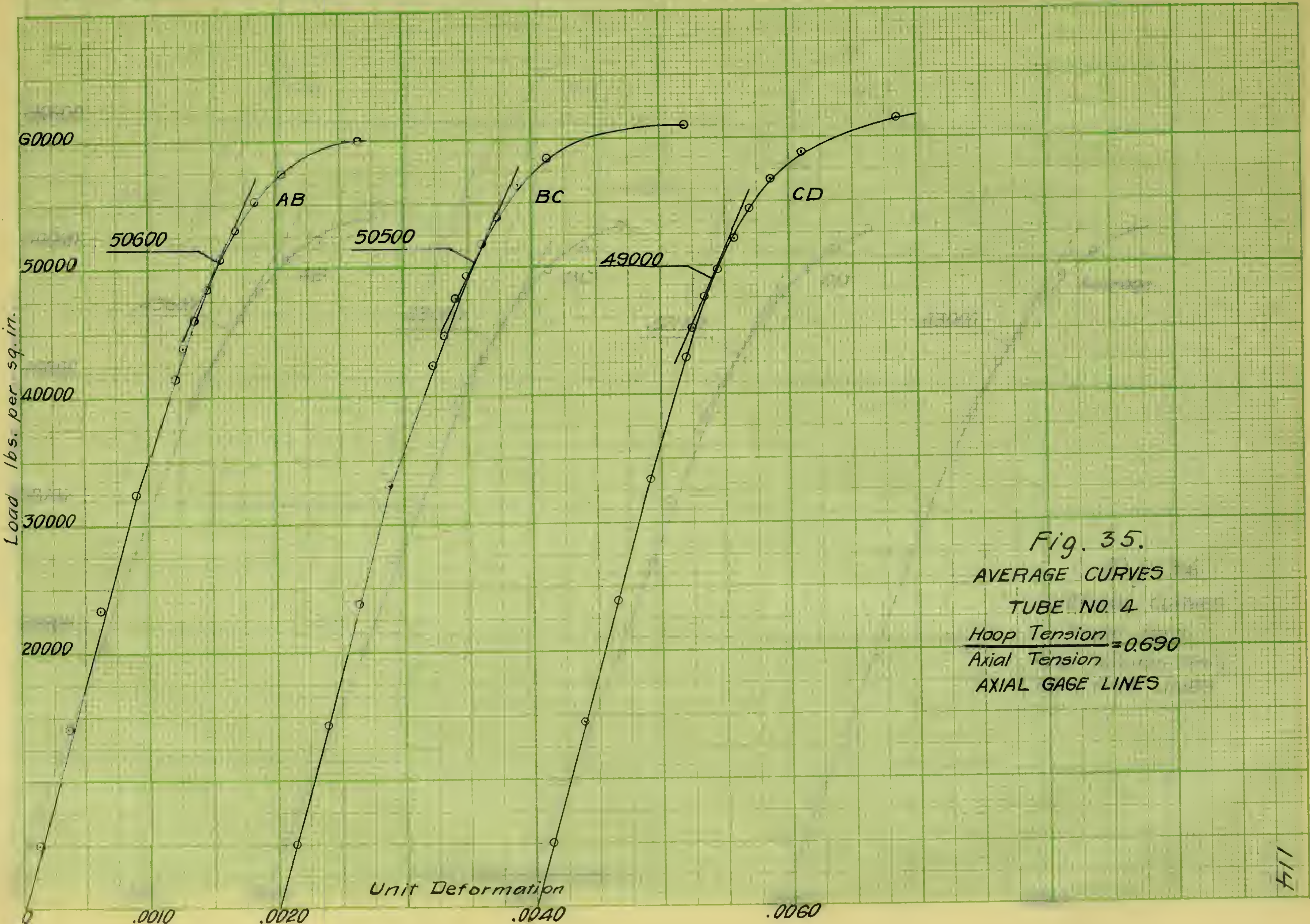


Fig. 35.
AVERAGE CURVES
TUBE NO. 4
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.690$
AXIAL GAGE LINES

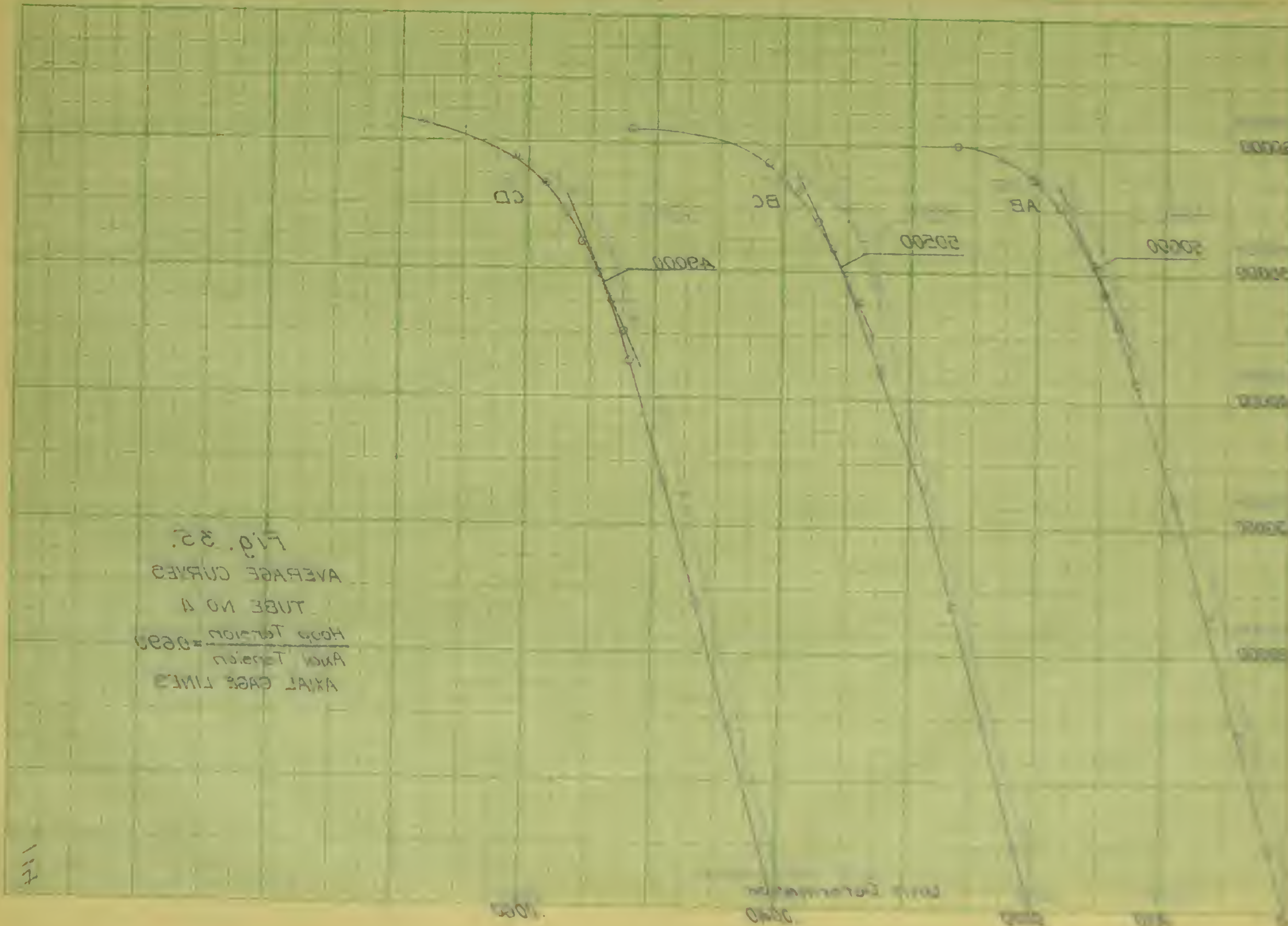


Fig. 35.
AVERAGE CURVES
TUBE NO 11
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.690$
AXIAL GAGE LINES

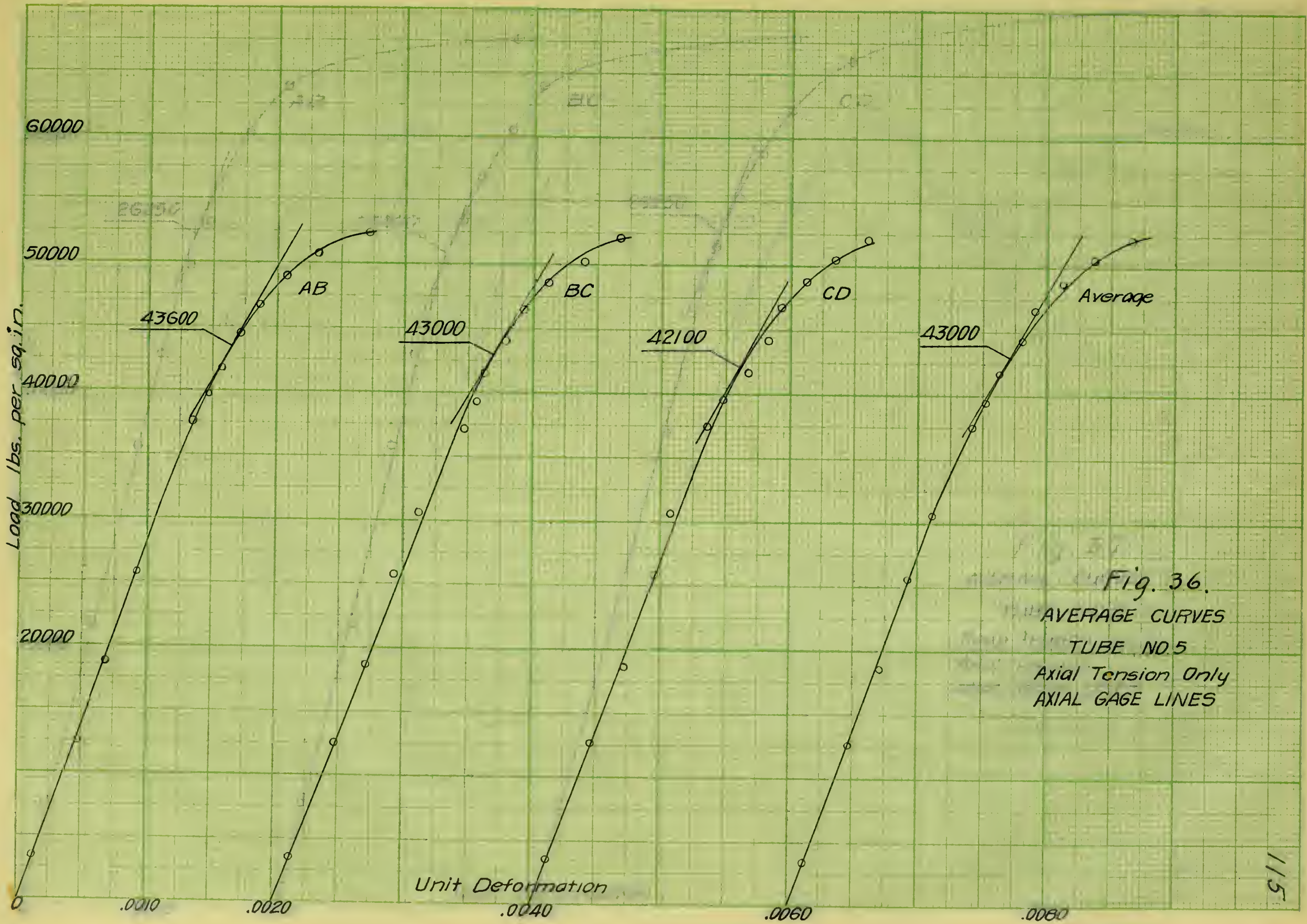


Fig. 36.
AVERAGE CURVES
TUBE NO. 5
Axial Tension Only
AXIAL GAGE LINES

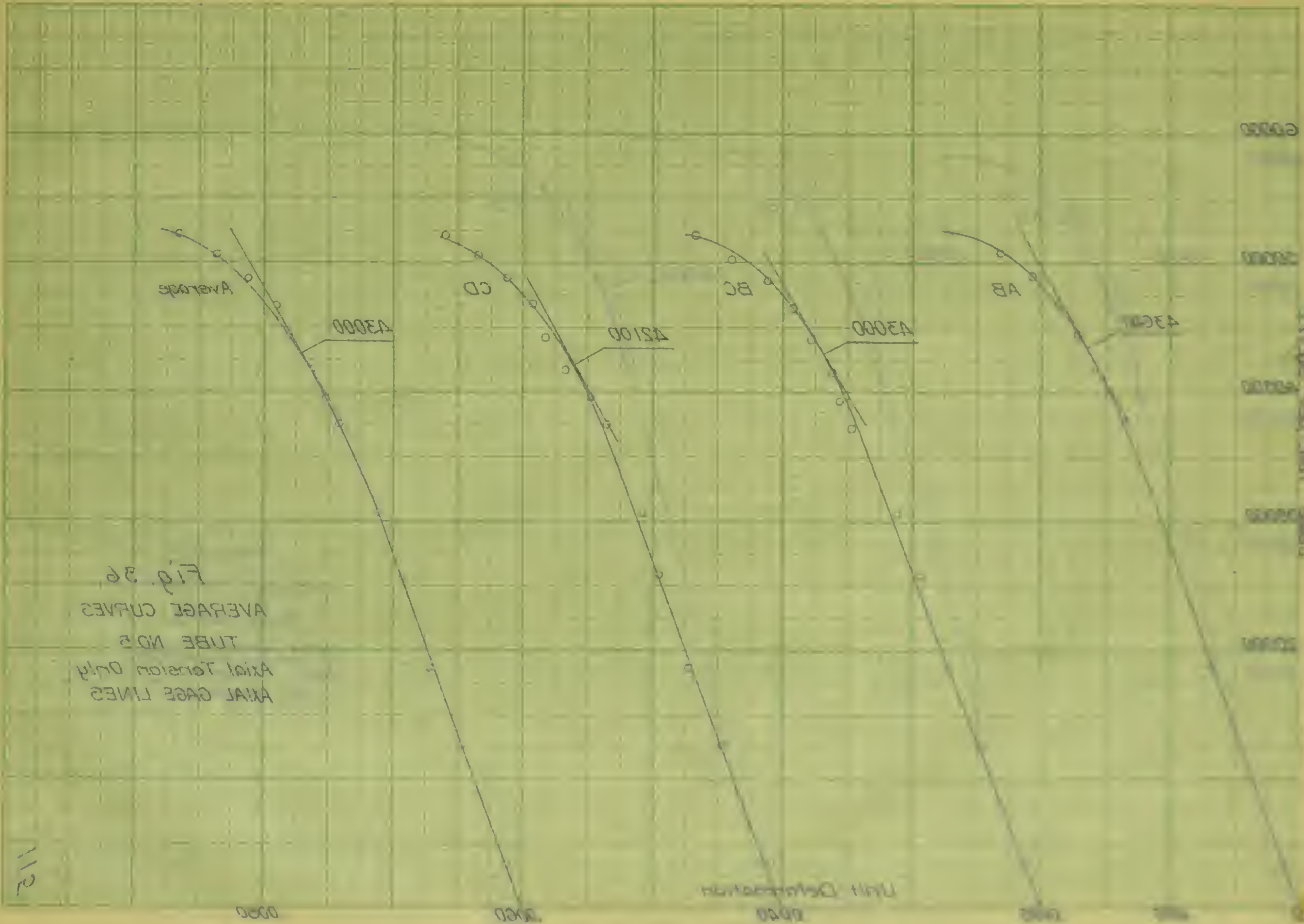


Fig. 36.
AVERAGE CURVES
TUBE NO. 5
Axial Tension Only
AXIAL GAGE LINES
Average
CD
BC
AB
43000
42100
43000
43600
60000
50000
40000
30000
20000
Unit Deformation
.0060
.0040

115

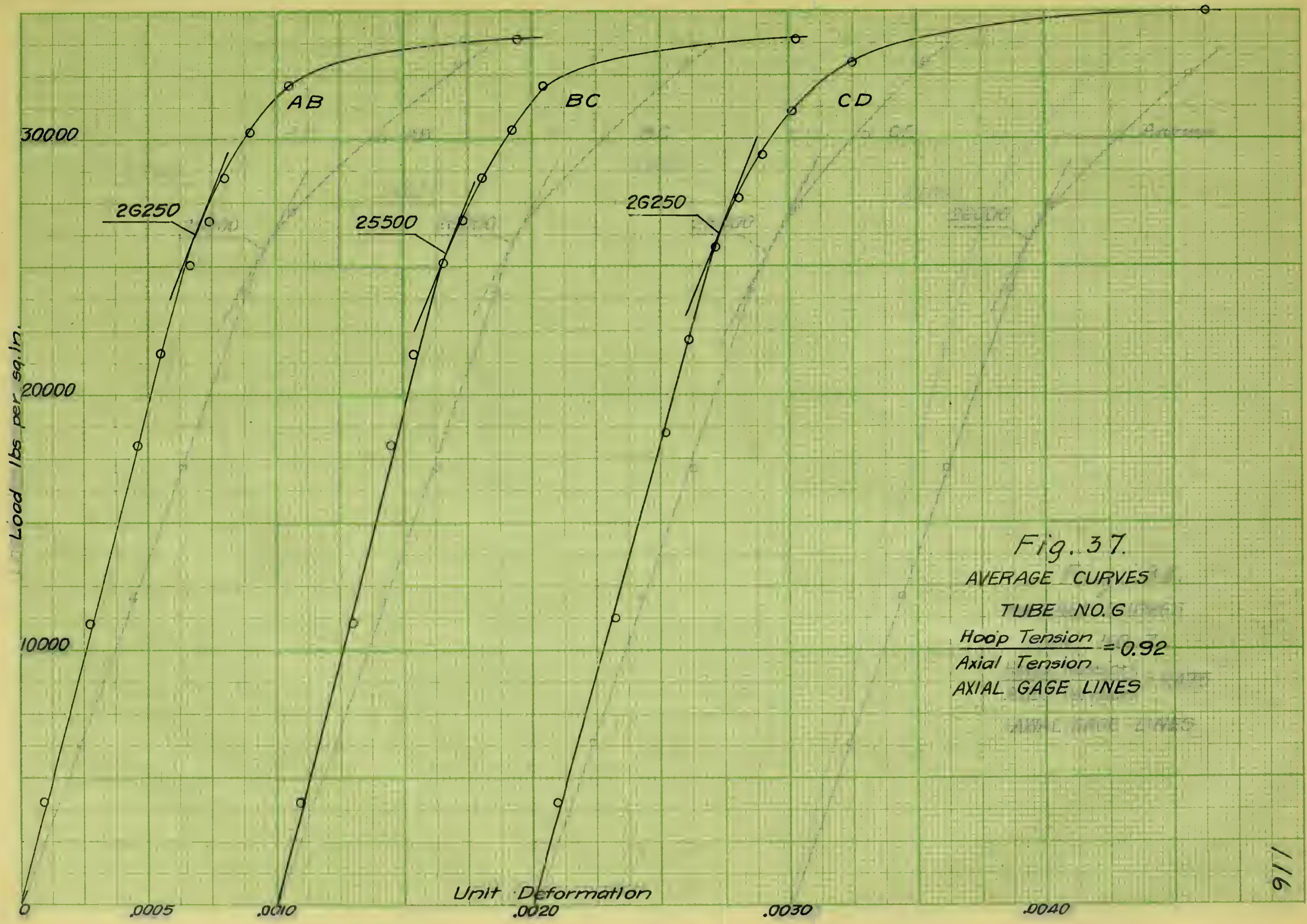


Fig. 37.
AVERAGE CURVES
TUBE NO. 6
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.92$
AXIAL GAGE LINES

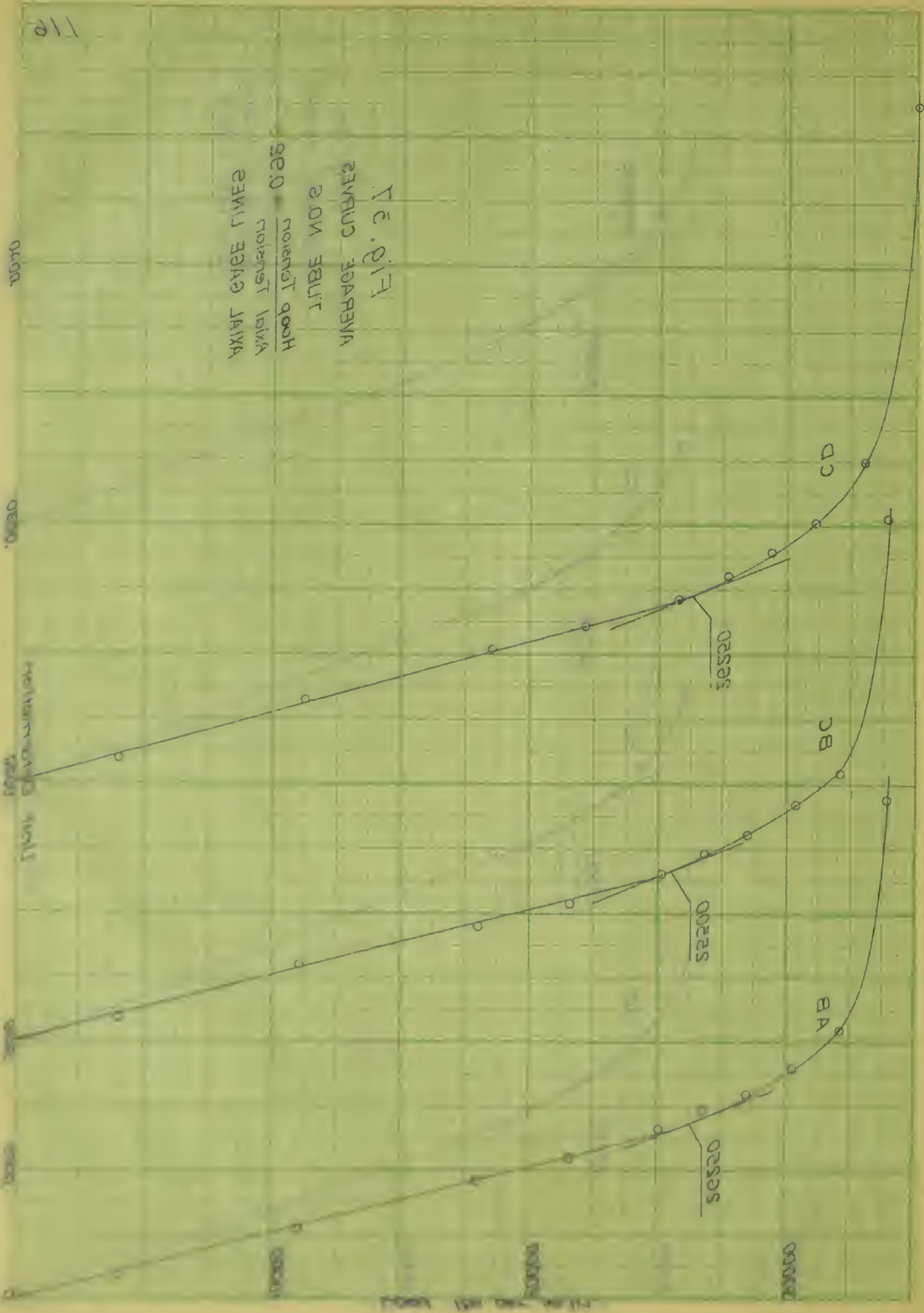


Fig. 3.7

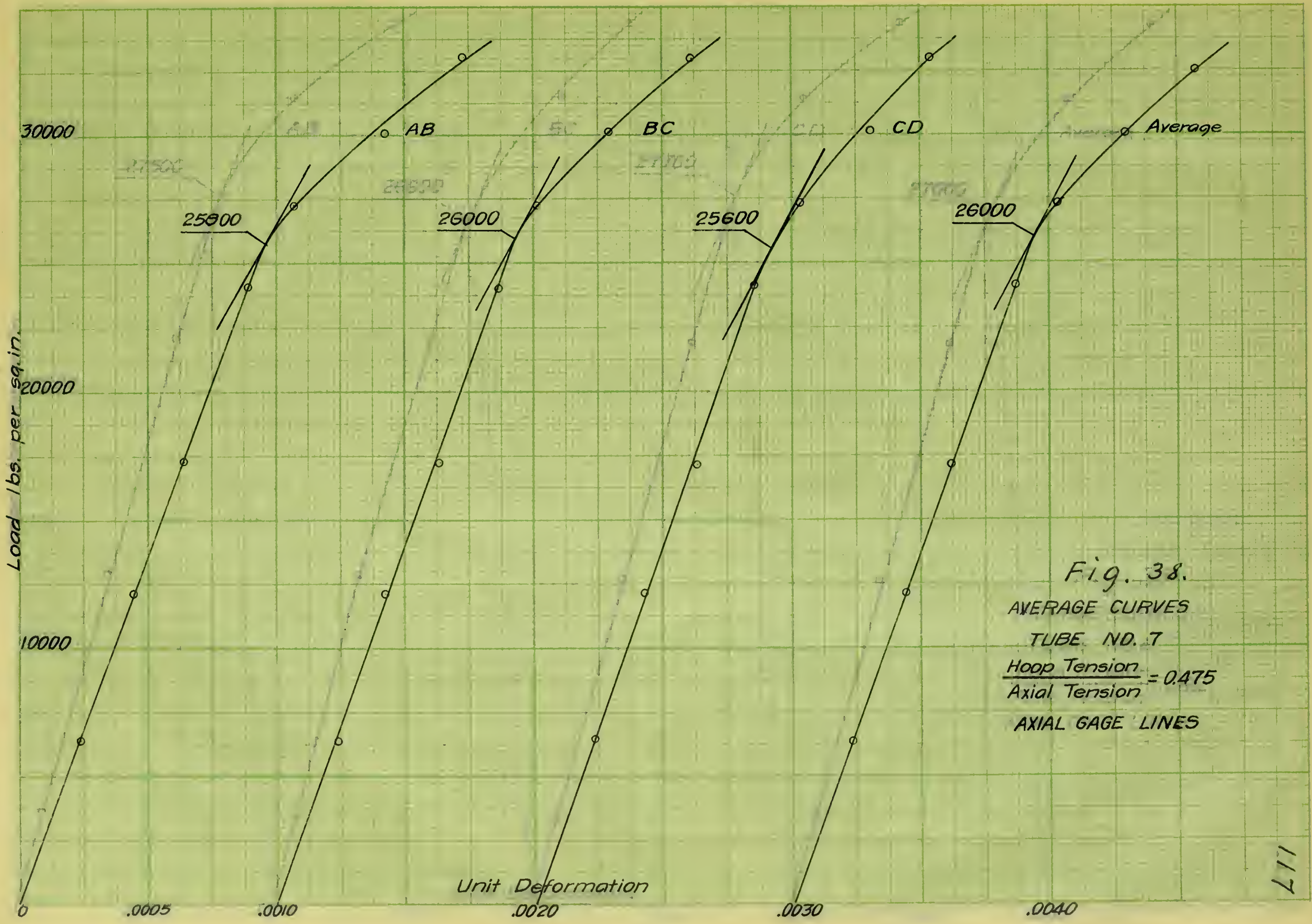
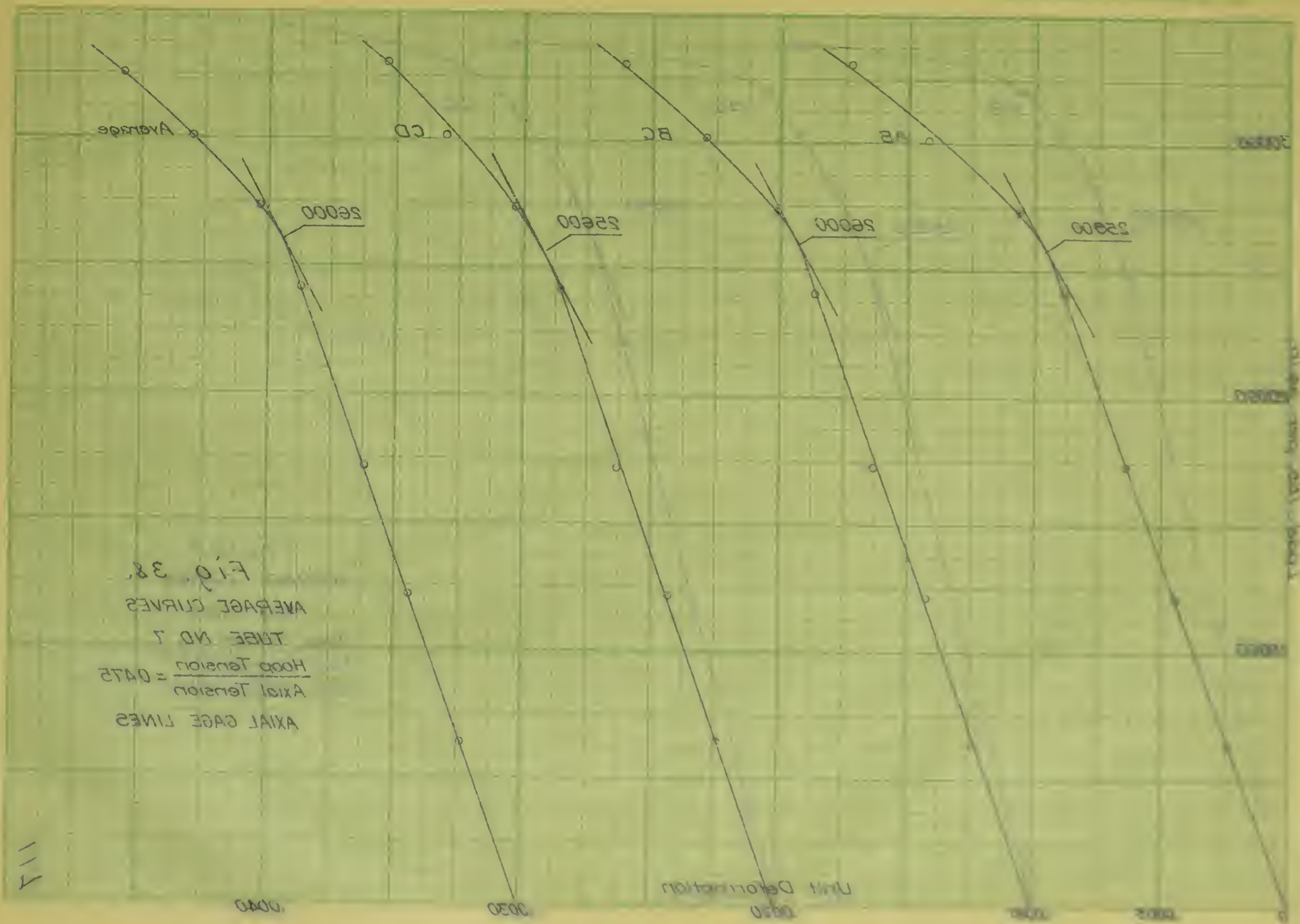


Fig. 38.
AVERAGE CURVES
TUBE NO. 7
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.475$
AXIAL GAGE LINES



117
Average
CD
BC
AB
26000
25600
26000
25600
AVERAGE CURVES
TUBE NO 7
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.475$
AXIAL GAGE LINES
Unit Deformation
.0040
.0030
.0020
.0005
0
30000
20000
10000

Fig. 38.

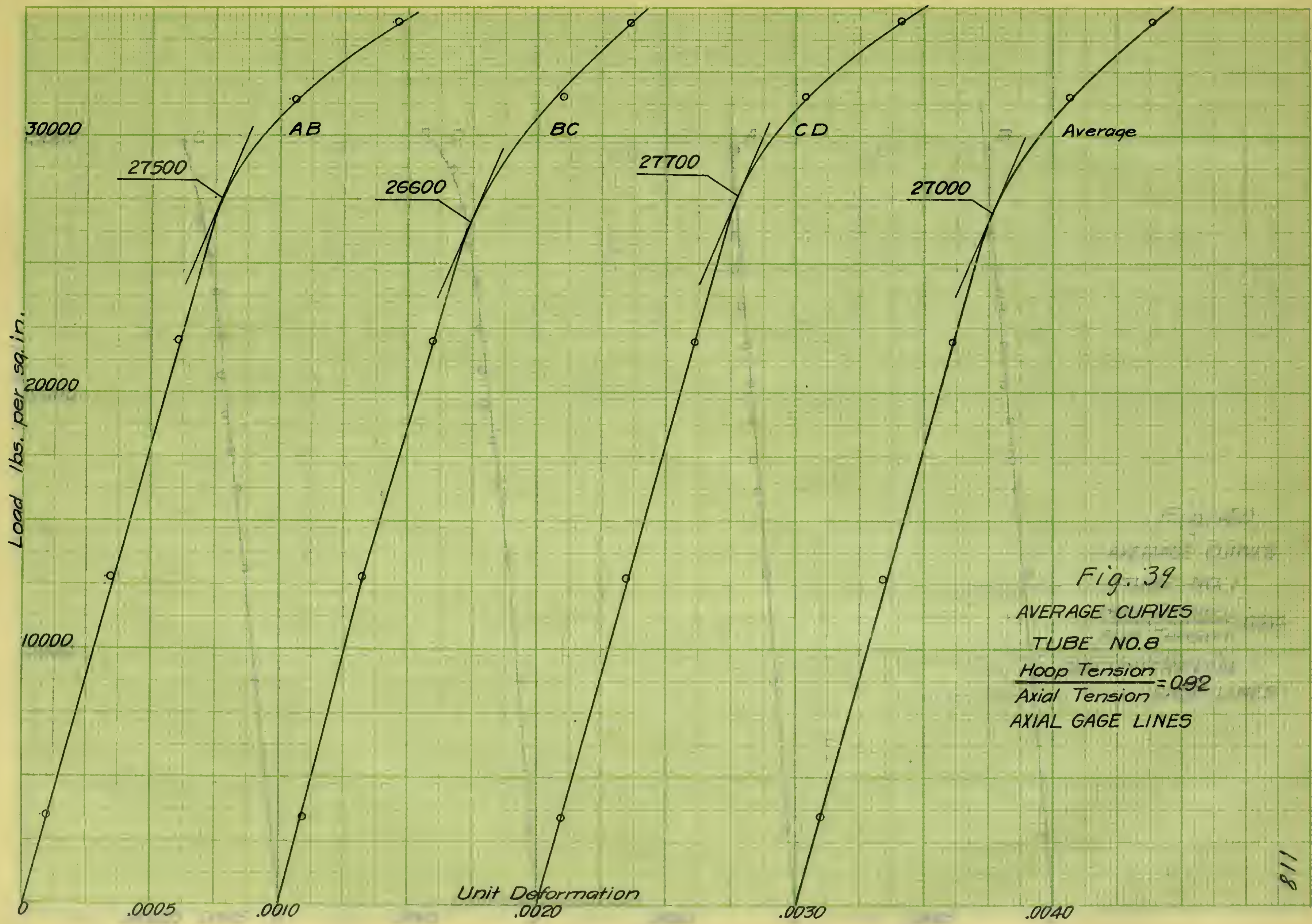


Fig. 39
AVERAGE CURVES
TUBE NO. 8
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.92$
AXIAL GAGE LINES

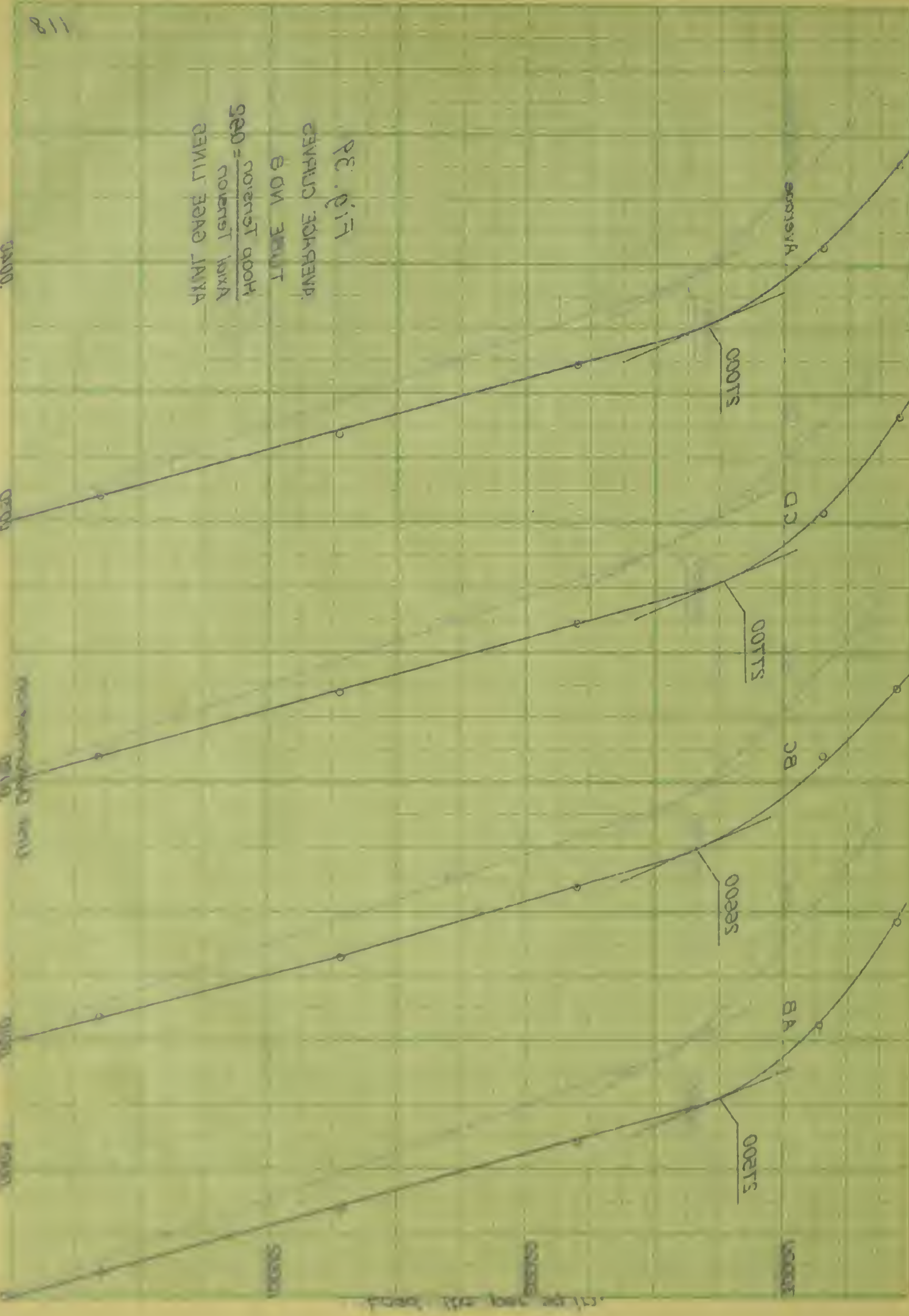


Fig. 39
AVERAGE CURVES
TUBE NO 8
Hoop Tension/Axial Tension = .92
AXIAL GAGE LINES
27500
26600
27700
27000
AB
BC
CD
Average
Unit Deformation
Load

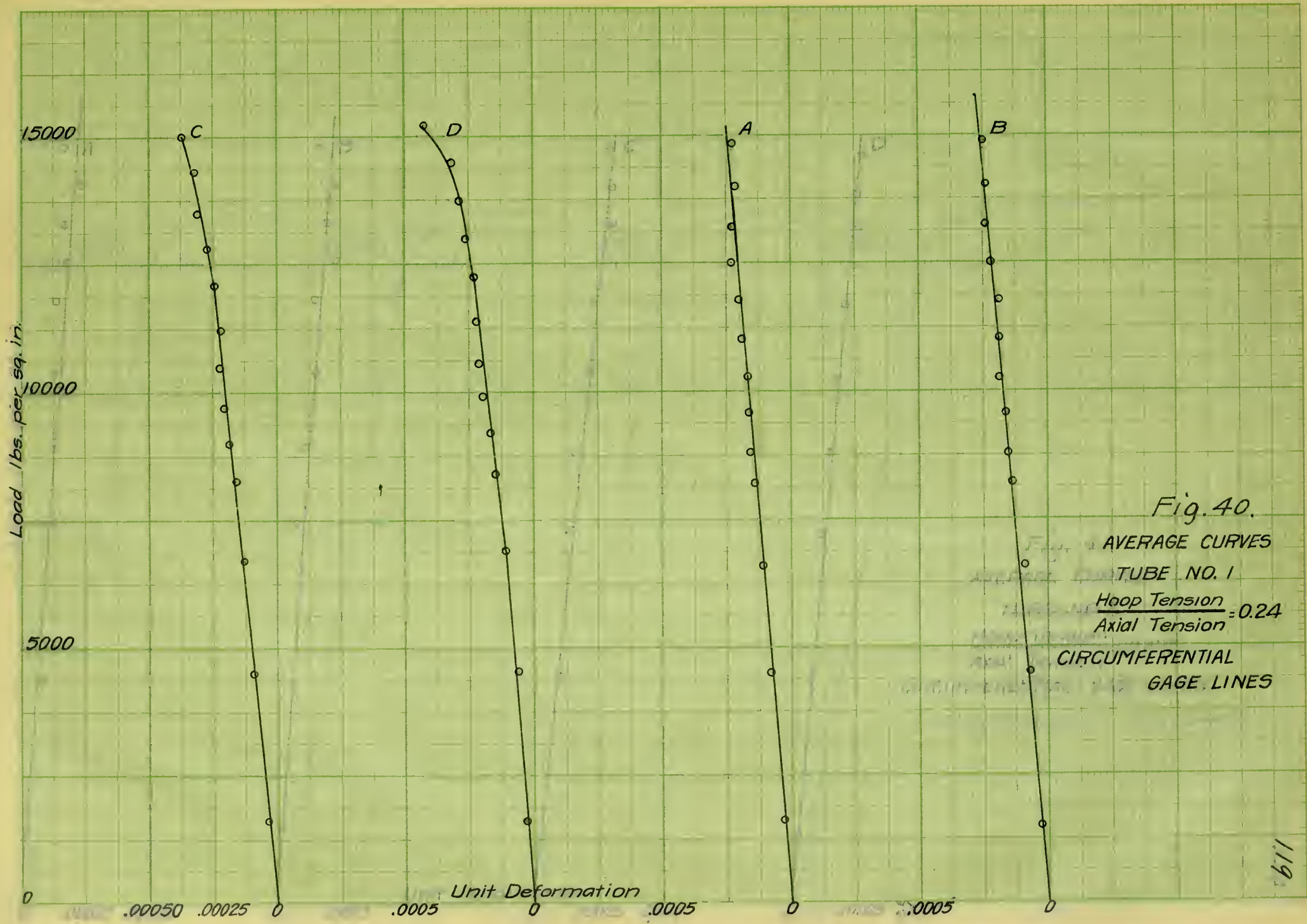


Fig. 40.
AVERAGE CURVES
TUBE NO. 1
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.24$
CIRCUMFERENTIAL
GAGE LINES

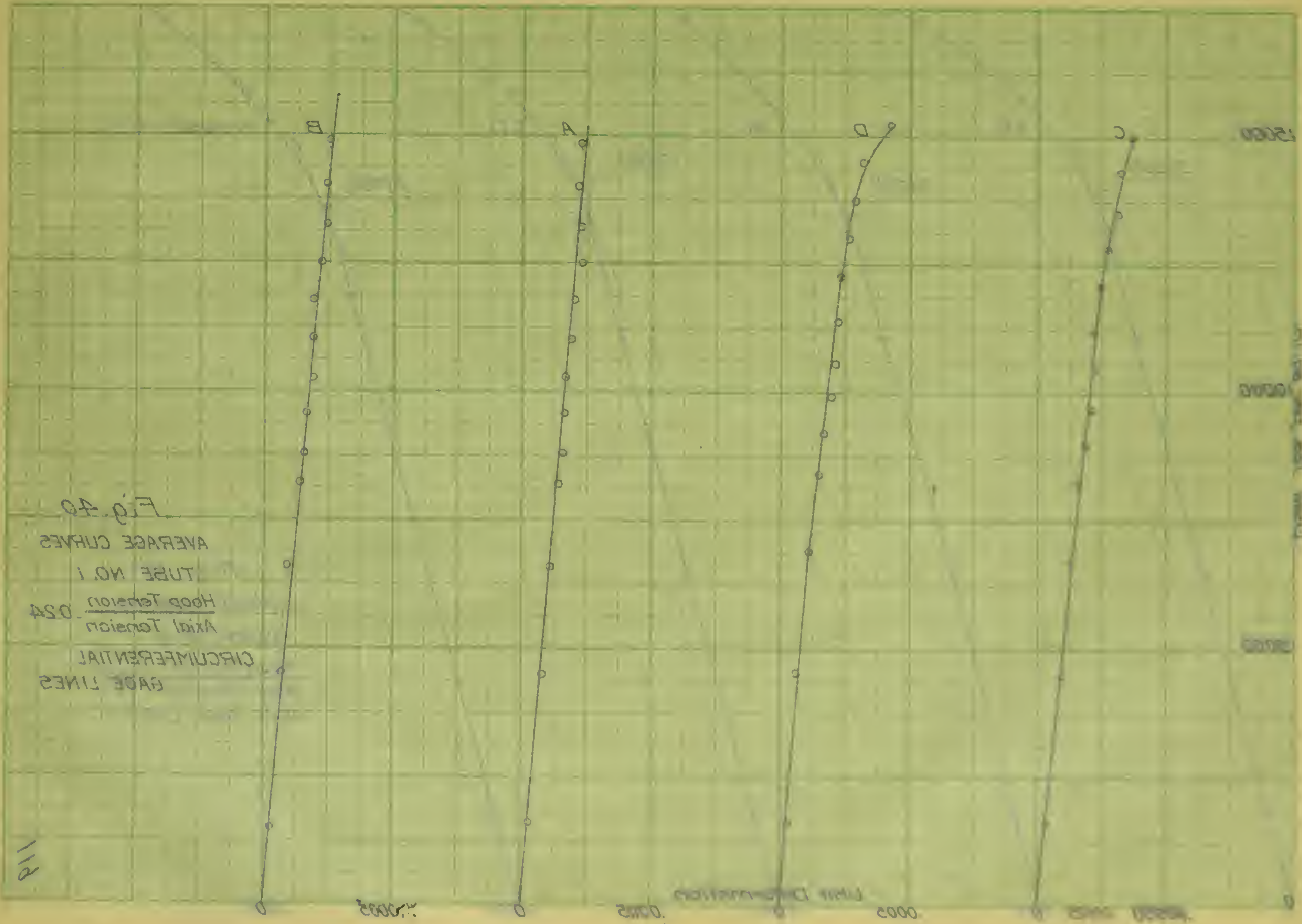


Fig. 40
AVERAGE CURVES
TUBE NO. 1
Hoop Tension / Axial Tension = 0.24
CIRCUMFERENTIAL
GAGE LINES
A
B
C
D
0
.0005
5000
10000
15000
Unit Deformation
119

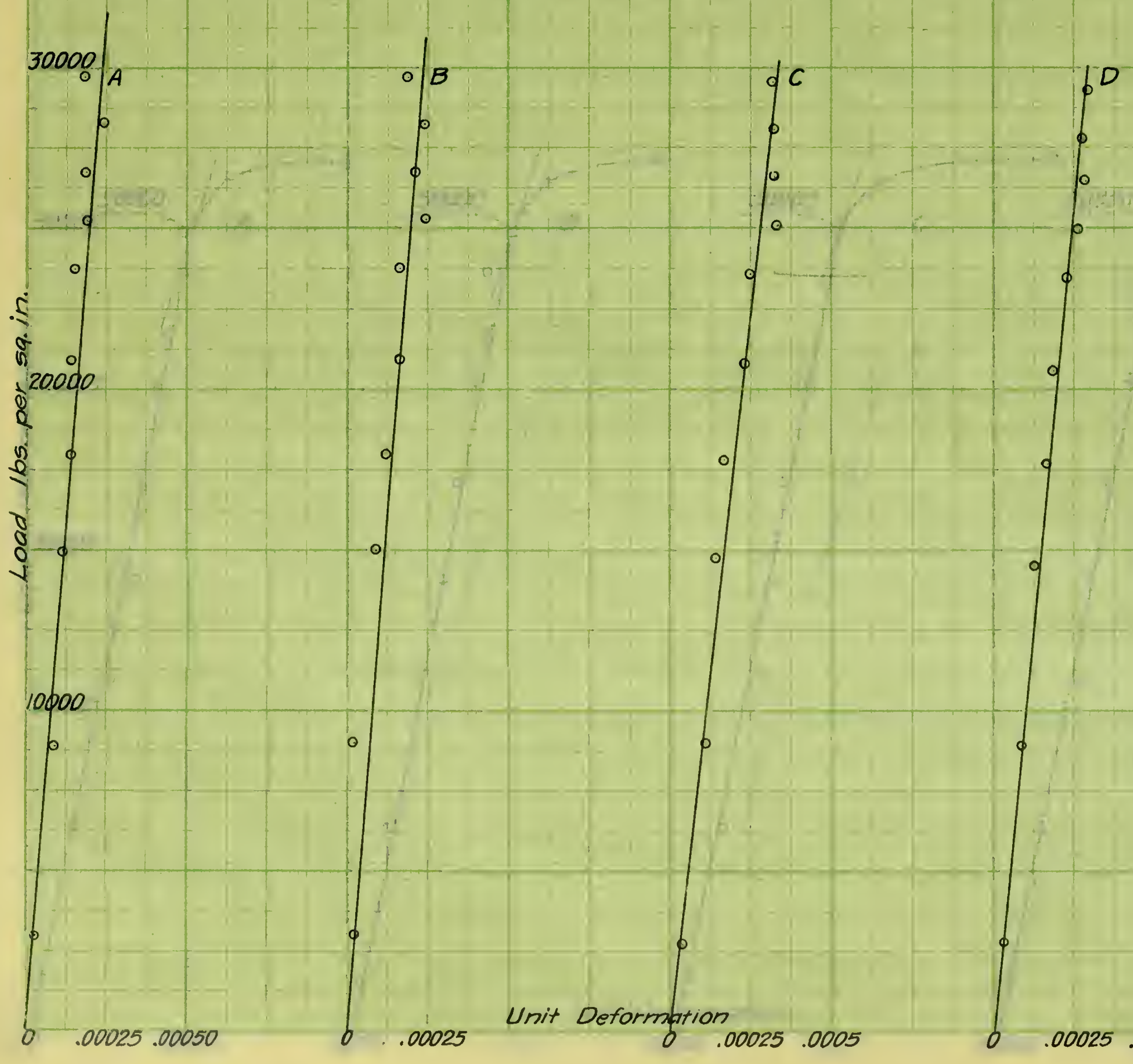


Fig. 41.

AVERAGE CURVES

TUBE NO. 2

$$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.475$$

CIRCUMFERENTIAL GAGE LINES

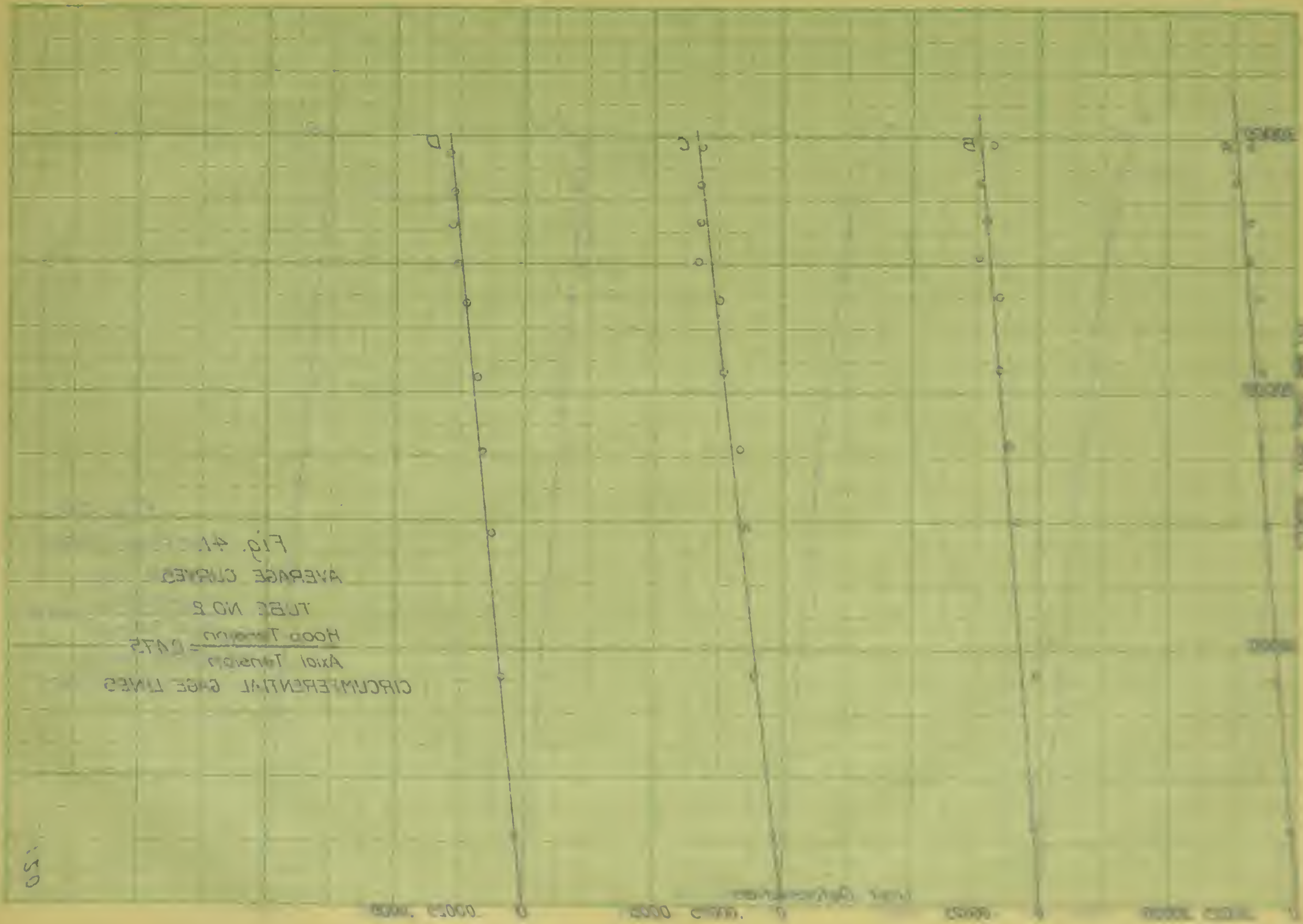


Fig. 41.
AVERAGE CURVES
TUBE NO 2
Hoop Tension / Axial Tension = .6475
CIRCUMFERENTIAL GAGE LINES
D
C
B
A

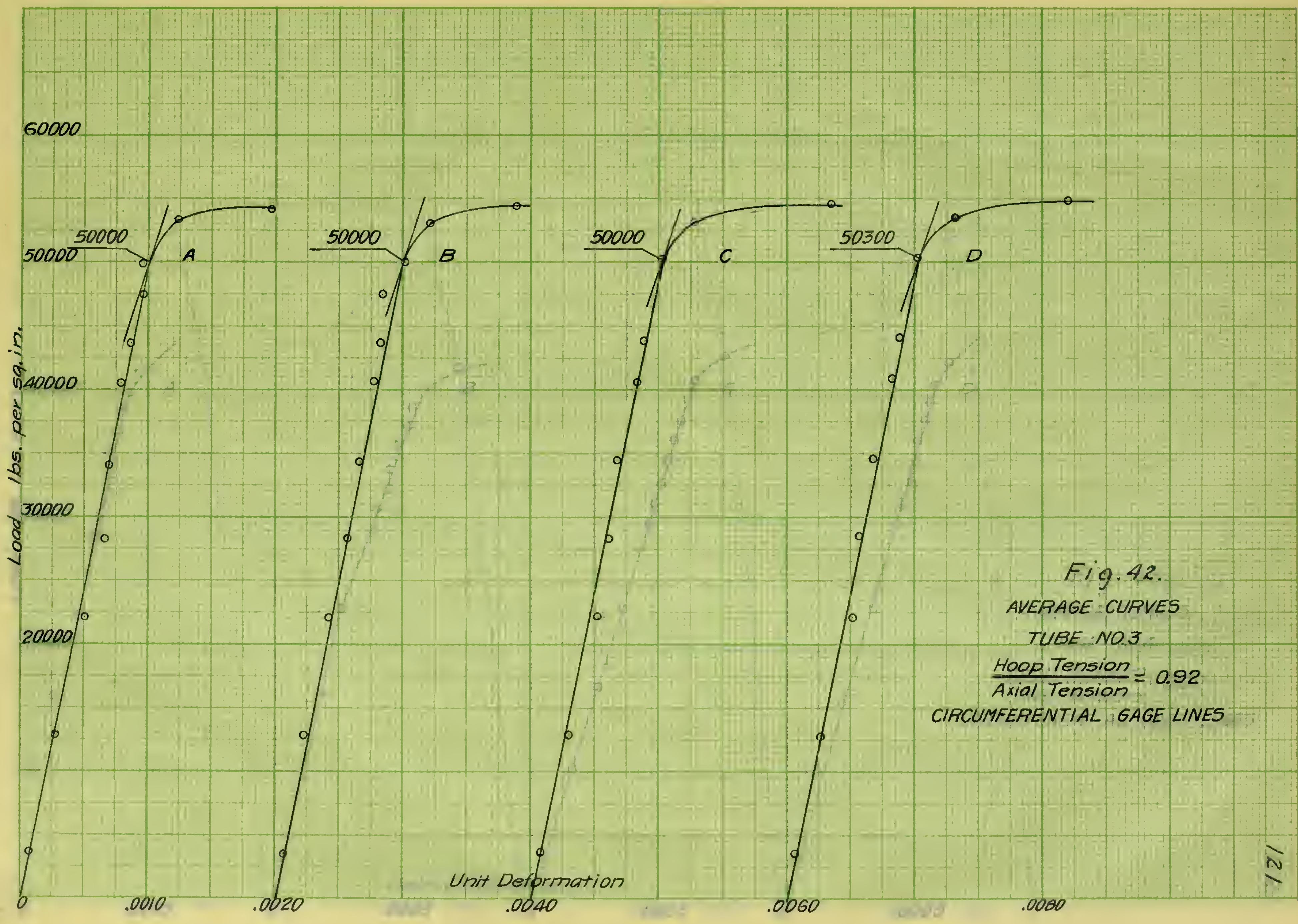


Fig. 42.

AVERAGE CURVES

TUBE NO. 3

$\dfrac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.92$

CIRCUMFERENTIAL GAGE LINES

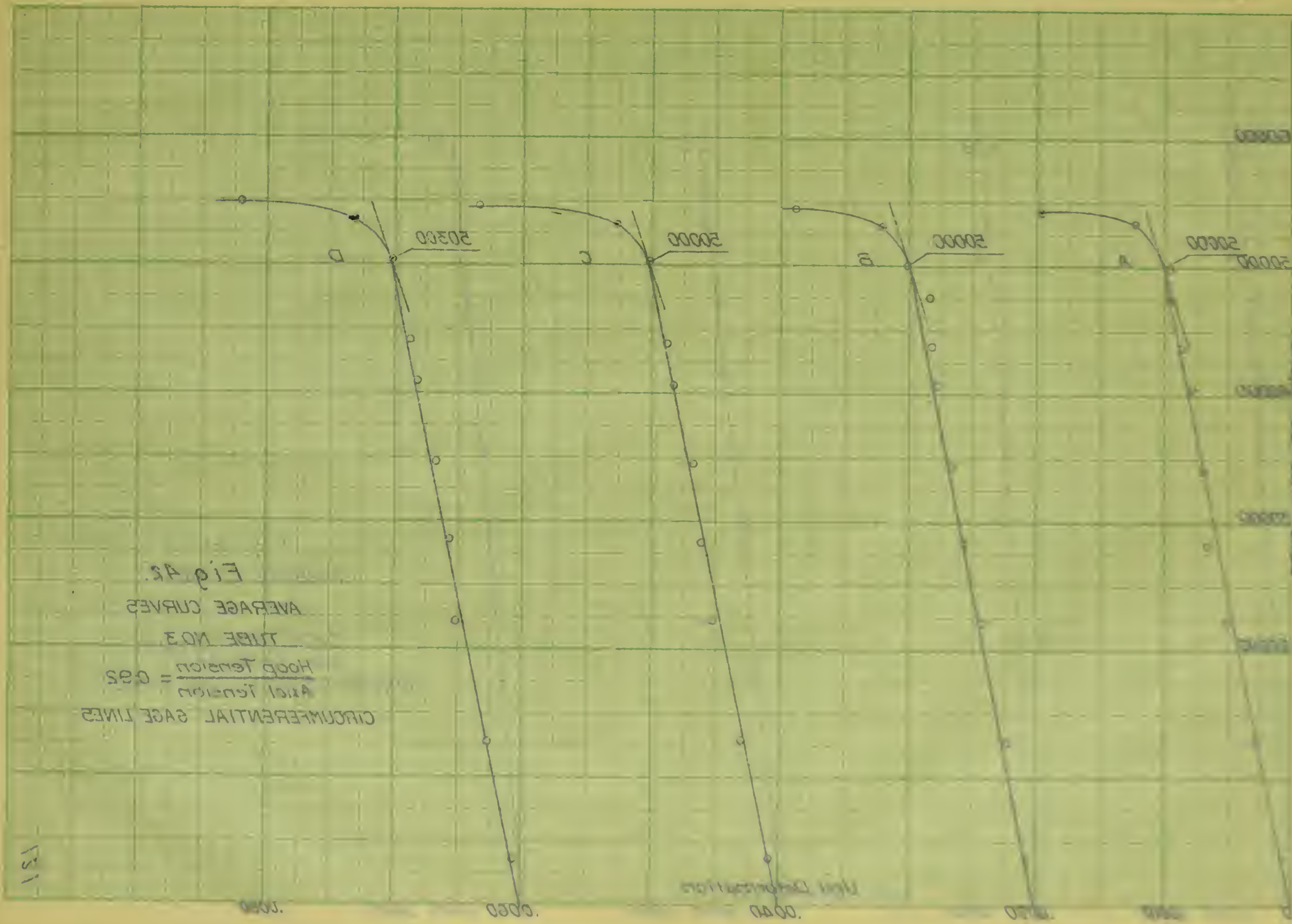


Fig. 42.
AVERAGE CURVES
TUBE NO.3
Hoop Tension / Axial Tension = 0.92
CIRCUMFERENTIAL GAGE LINES
50300
50000
50000
50000
A
B
C
D
.0000
.0020
.0040
.0060
.0080
Unit Deformations

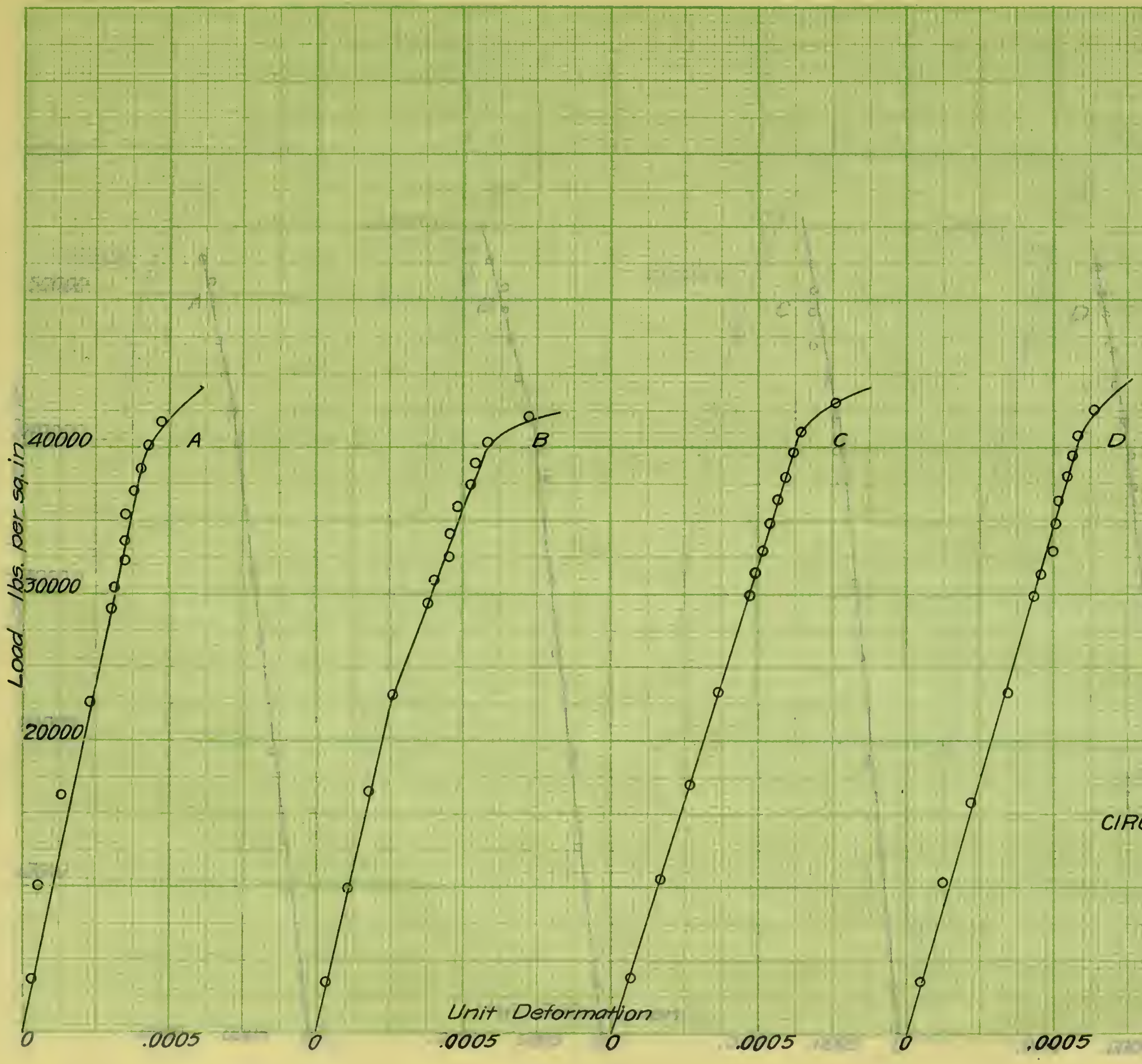


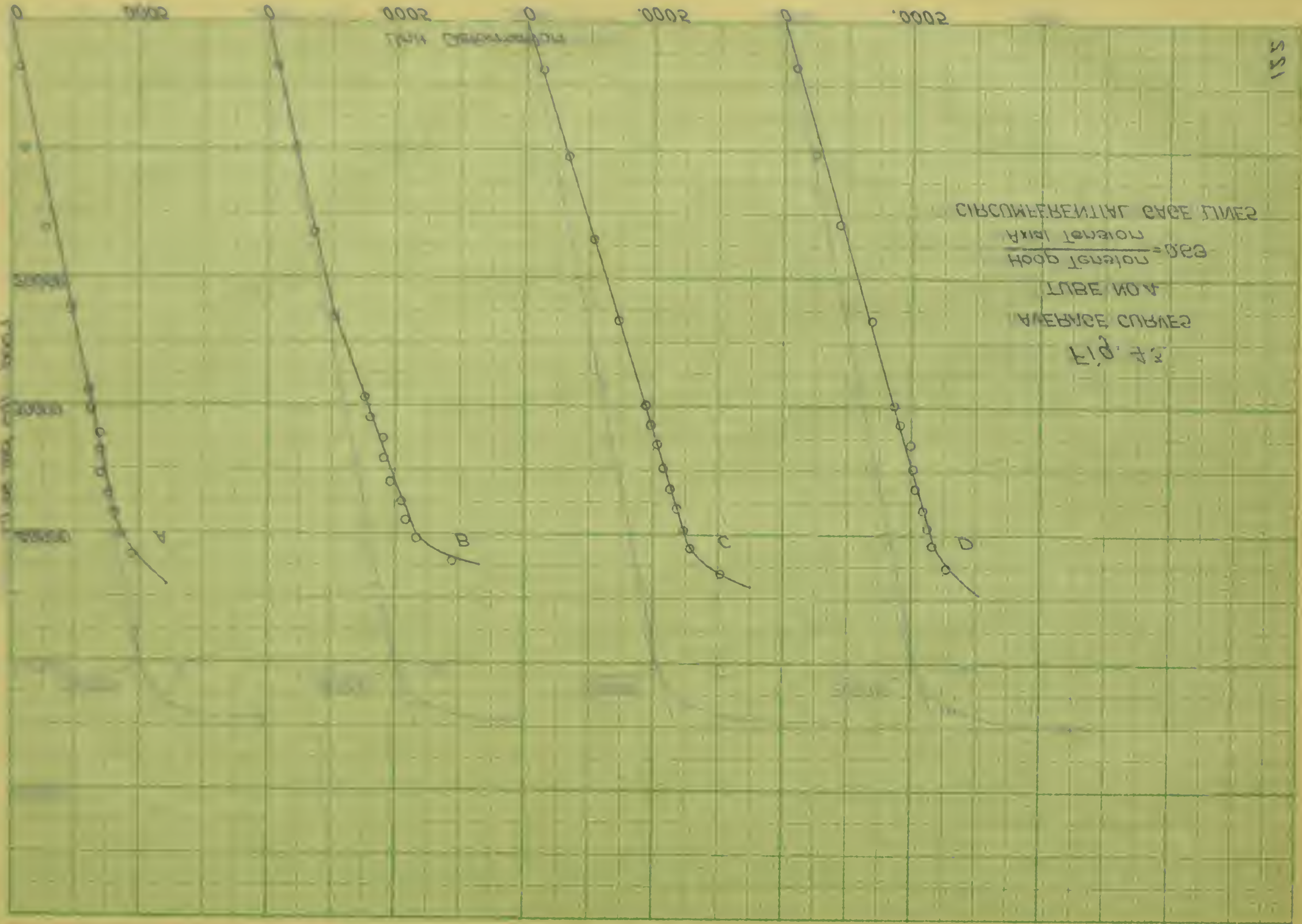
Fig. 43.

AVERAGE CURVES

TUBE NO.4

$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.69$

CIRCUMFERENTIAL GAGE LINES



CIRCUMFERENTIAL GAGE LINES
Axial Tension / Hoop Tension
TUBE NO. 4
AVERAGE CURVES
A
B
C
D
0
.0002

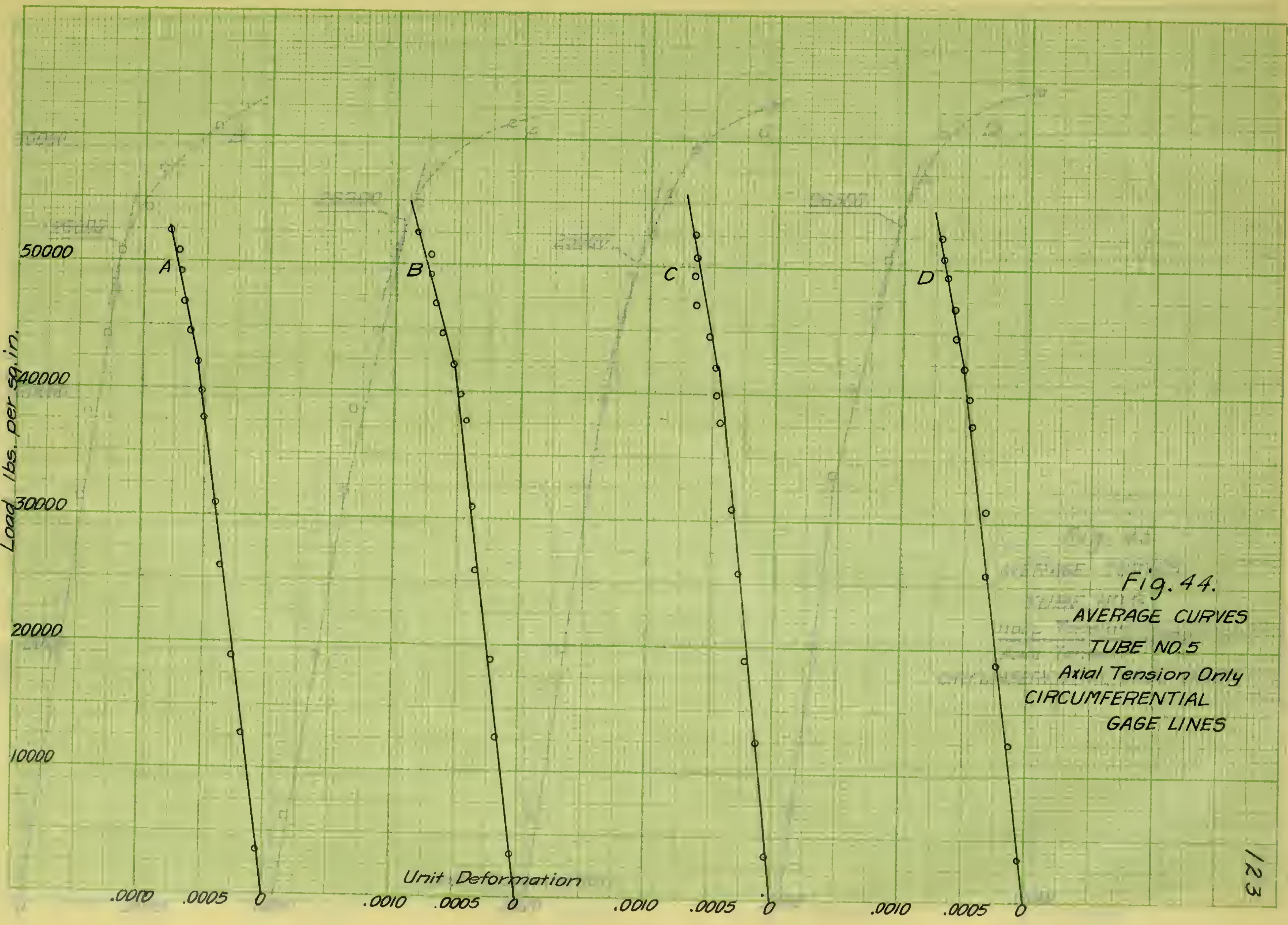


Fig. 44.
AVERAGE CURVES
TUBE NO. 5
Axial Tension Only
CIRCUMFERENTIAL
GAGE LINES

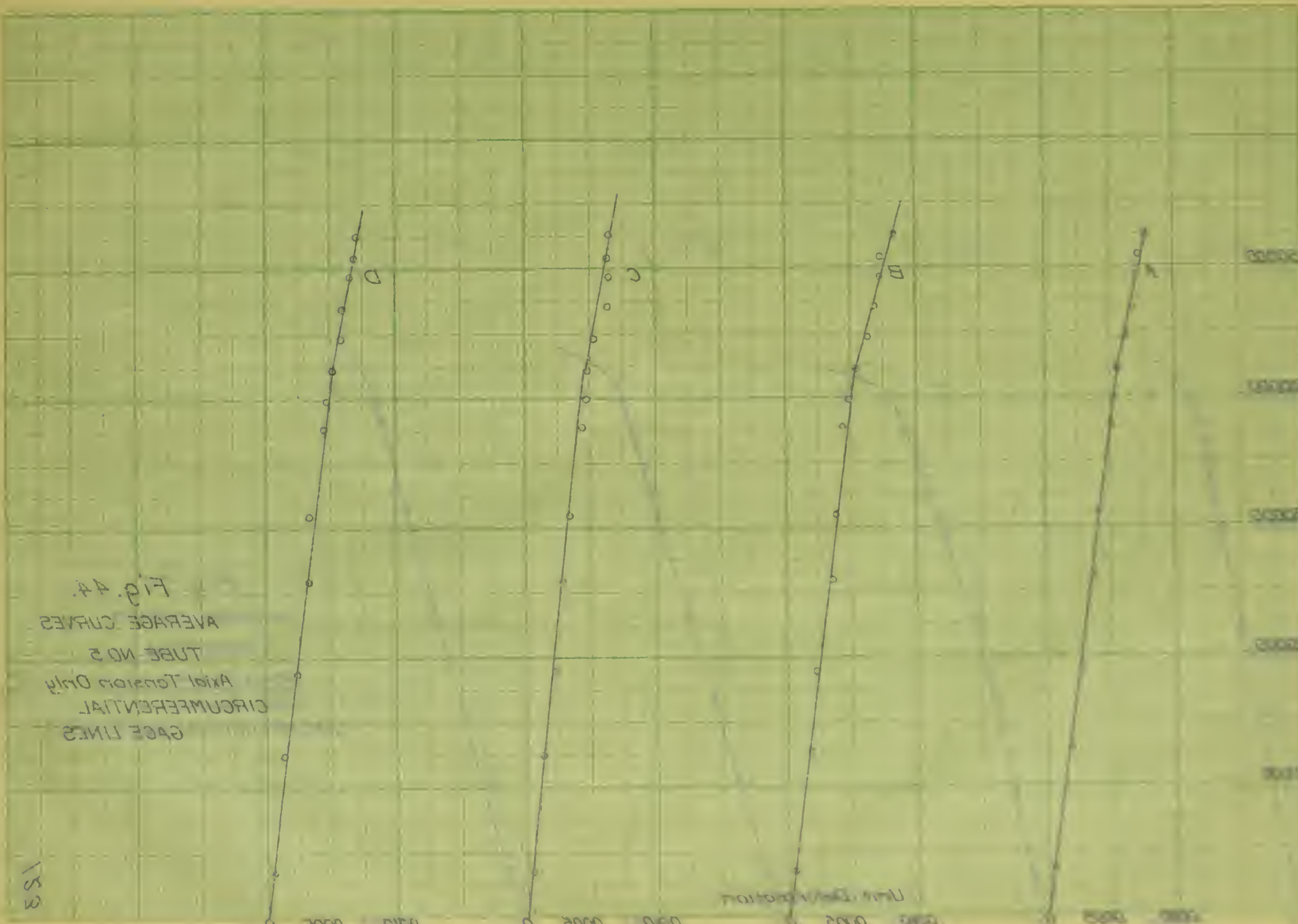


Fig. 44.
AVERAGE CURVES
TUBE NO 5
Axial Tension Only
CIRCUMFERENTIAL
GAGE LINES

123

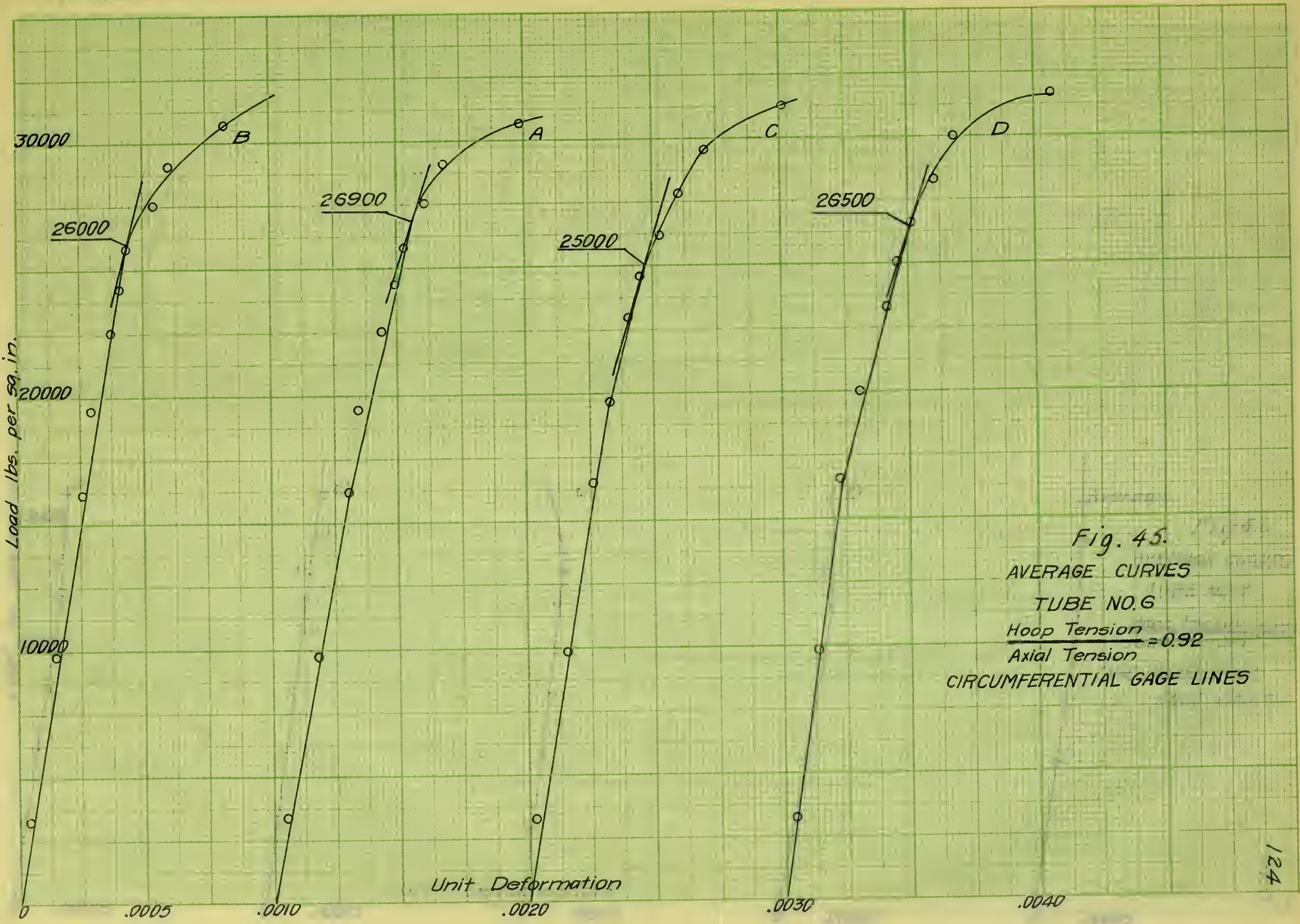
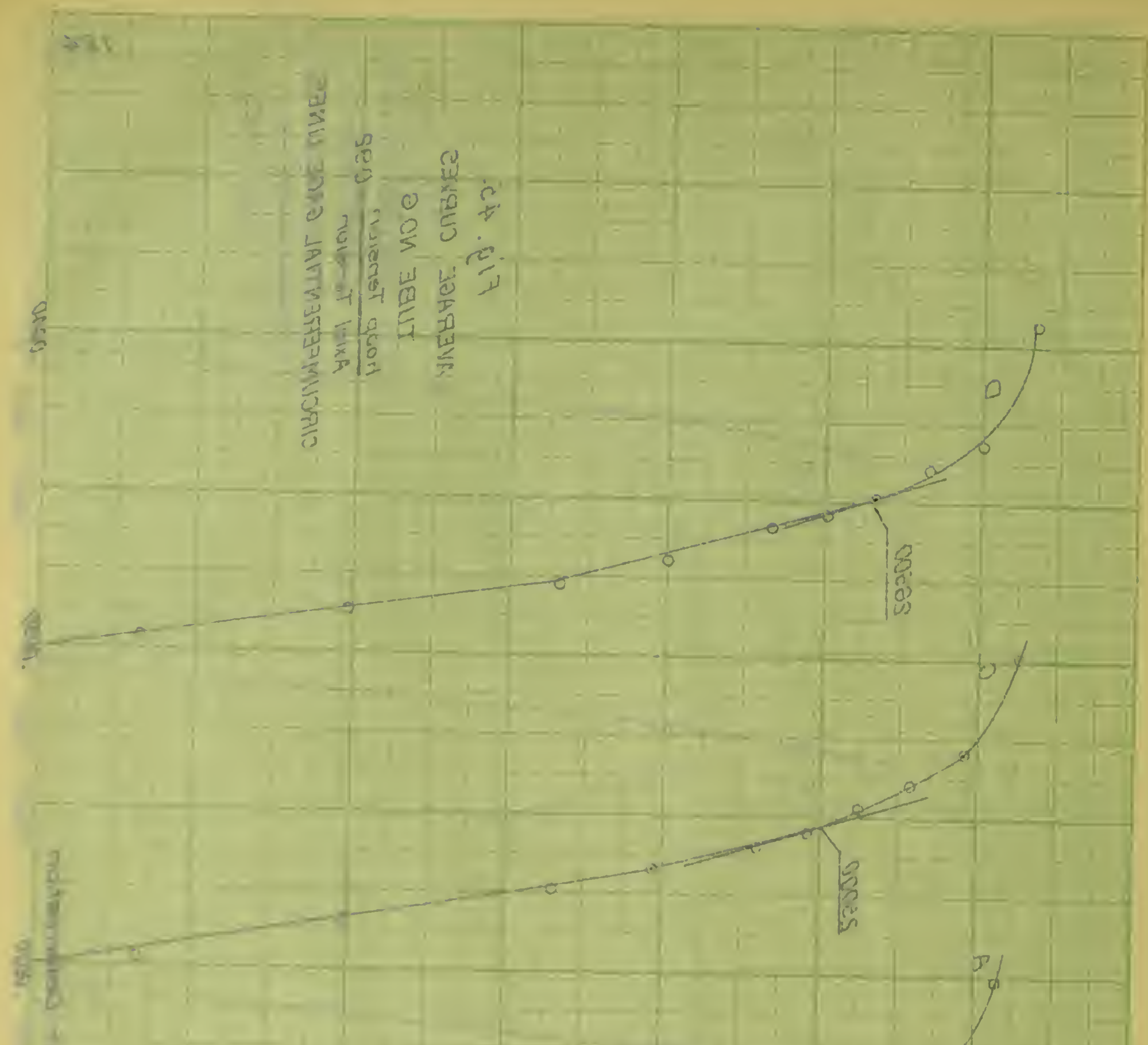


Fig. 45.
AVERAGE CURVES
TUBE NO. 6

$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.92$

CIRCUMFERENTIAL GAGE LINES



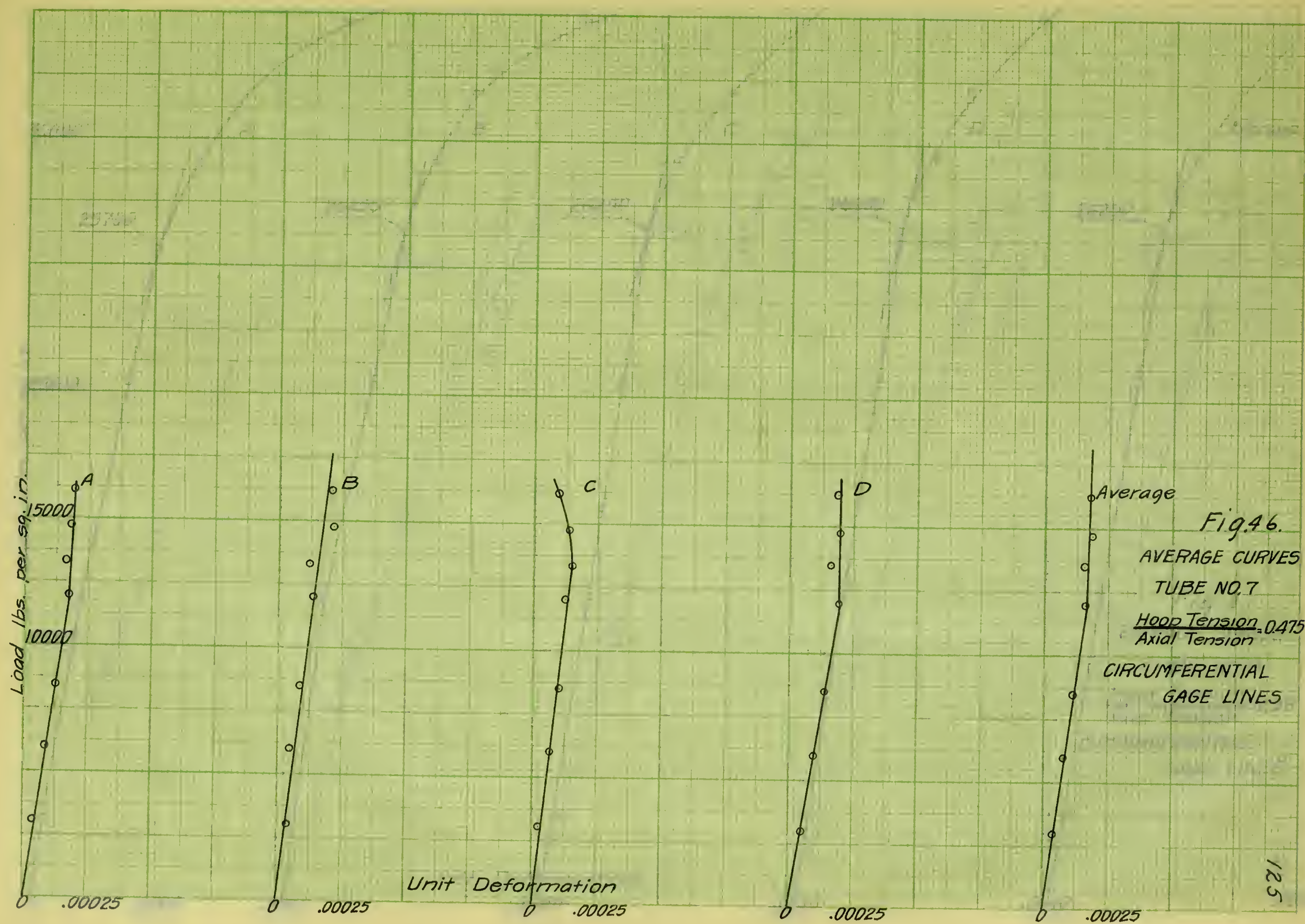
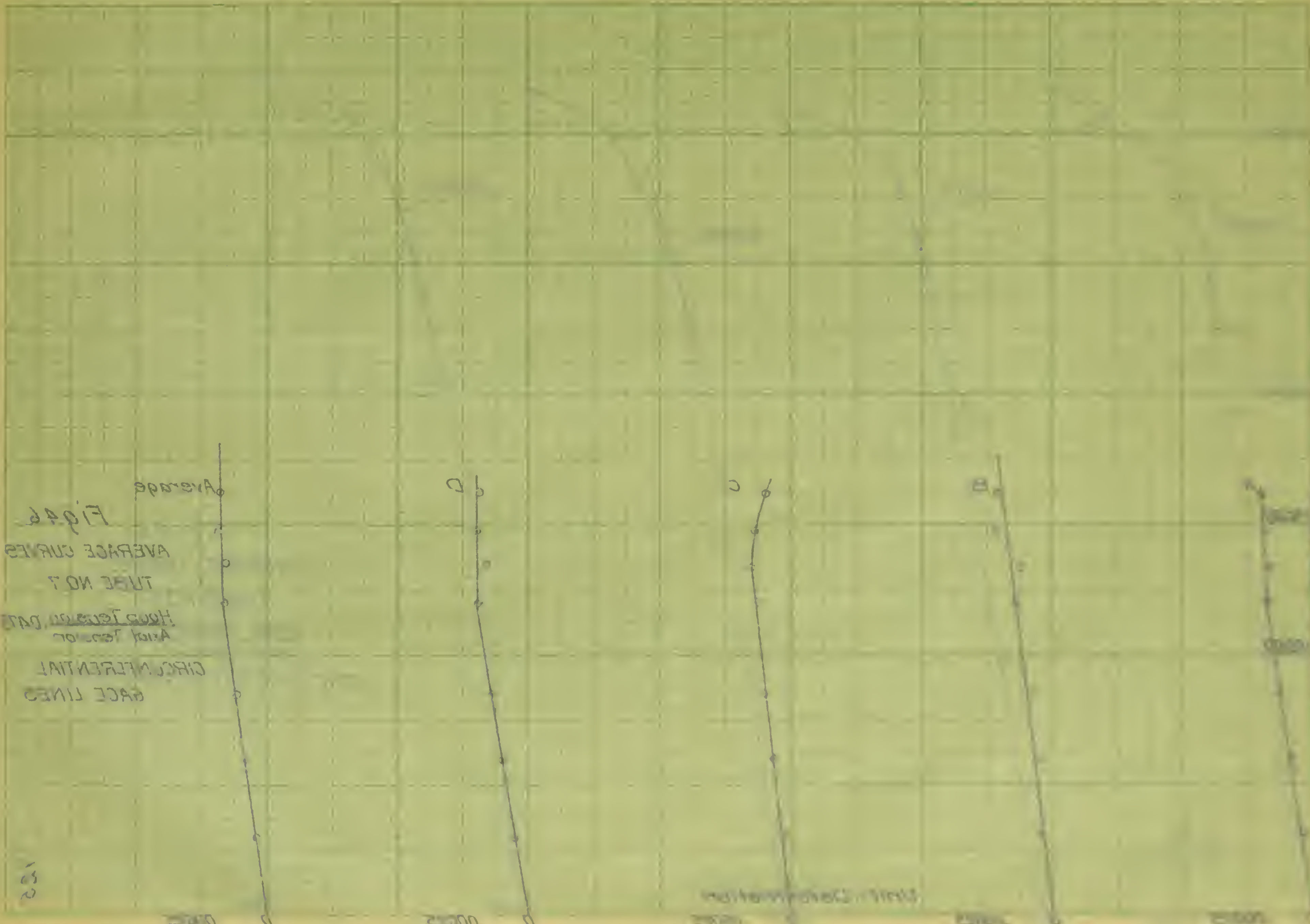


Fig. 46.
AVERAGE CURVES
TUBE NO. 7
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.475$
CIRCUMFERENTIAL
GAGE LINES


Average
AVERAGE CURVES
TUBE NO 7
CIRCUMFERENTIAL
GAGE LINES

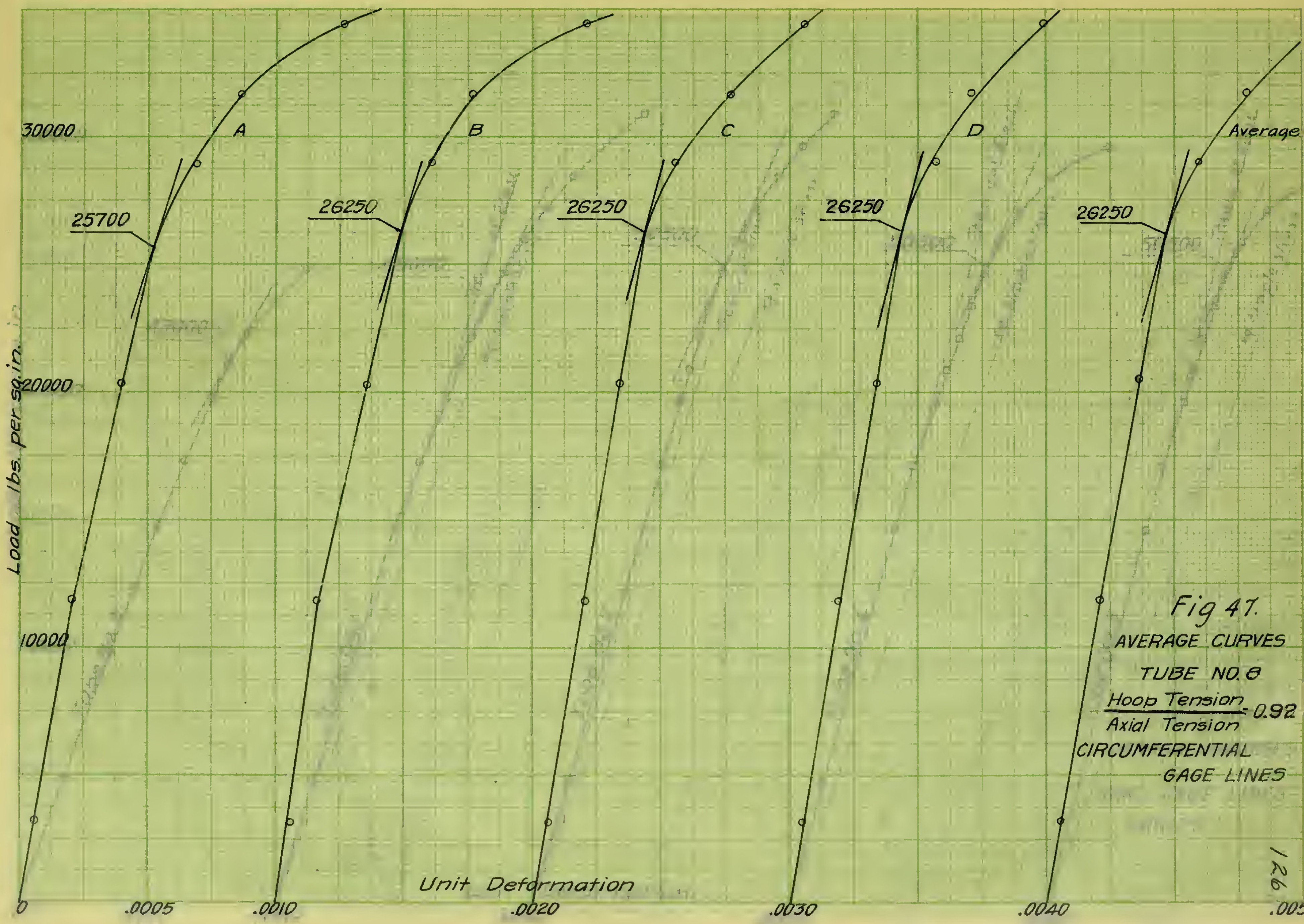


Fig 47.
AVERAGE CURVES
TUBE NO. 8
$\frac{\text{Hoop Tension}}{\text{Axial Tension}} = 0.92$
CIRCUMFERENTIAL
GAGE LINES

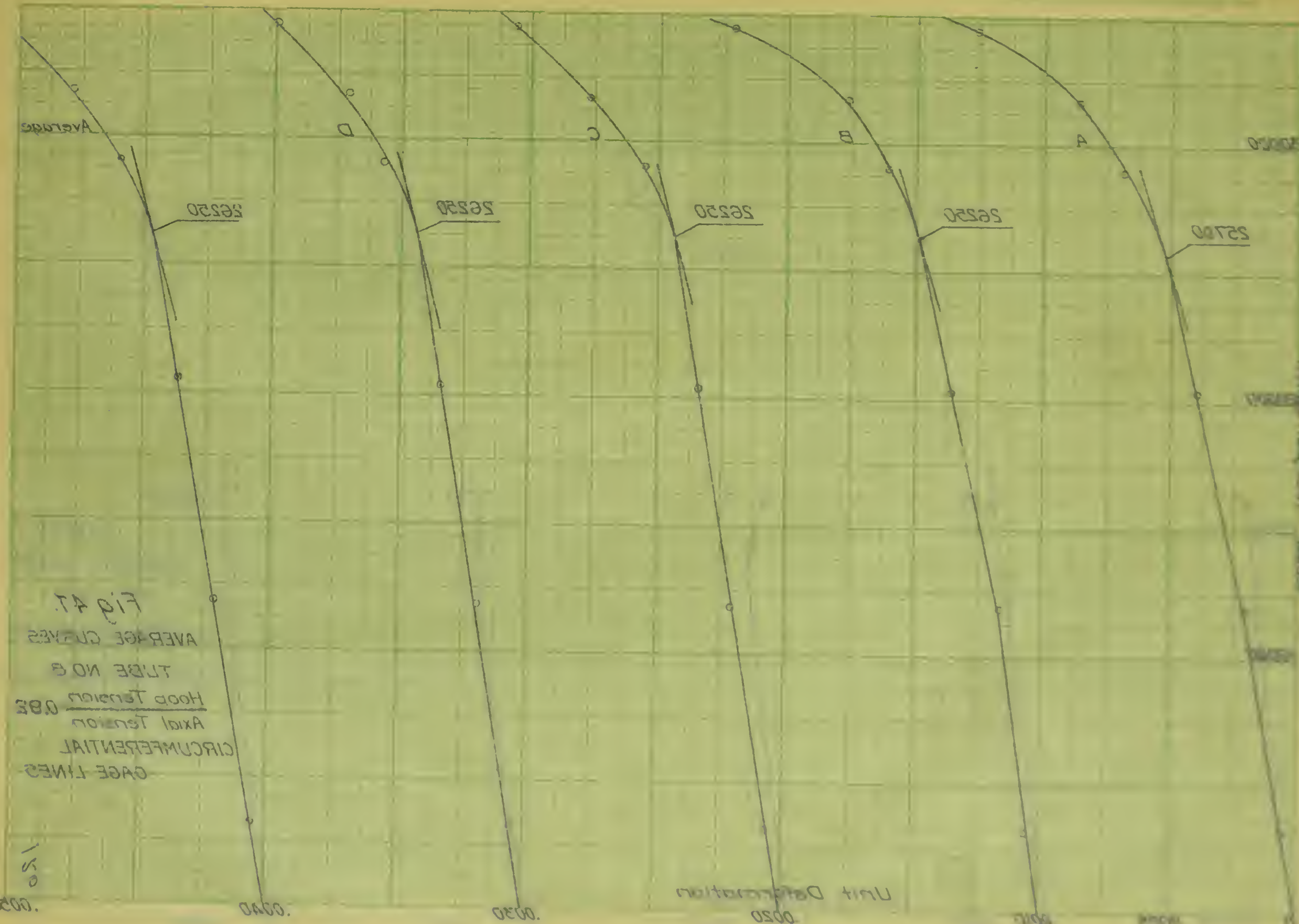


Fig 47.
AVERAGE CURVES
TUBE NO 8
$\frac{\text{Hoop Tension}}{\text{Axial Tension}}$ 0.82
CIRCUMFERENTIAL
GAGE LINES

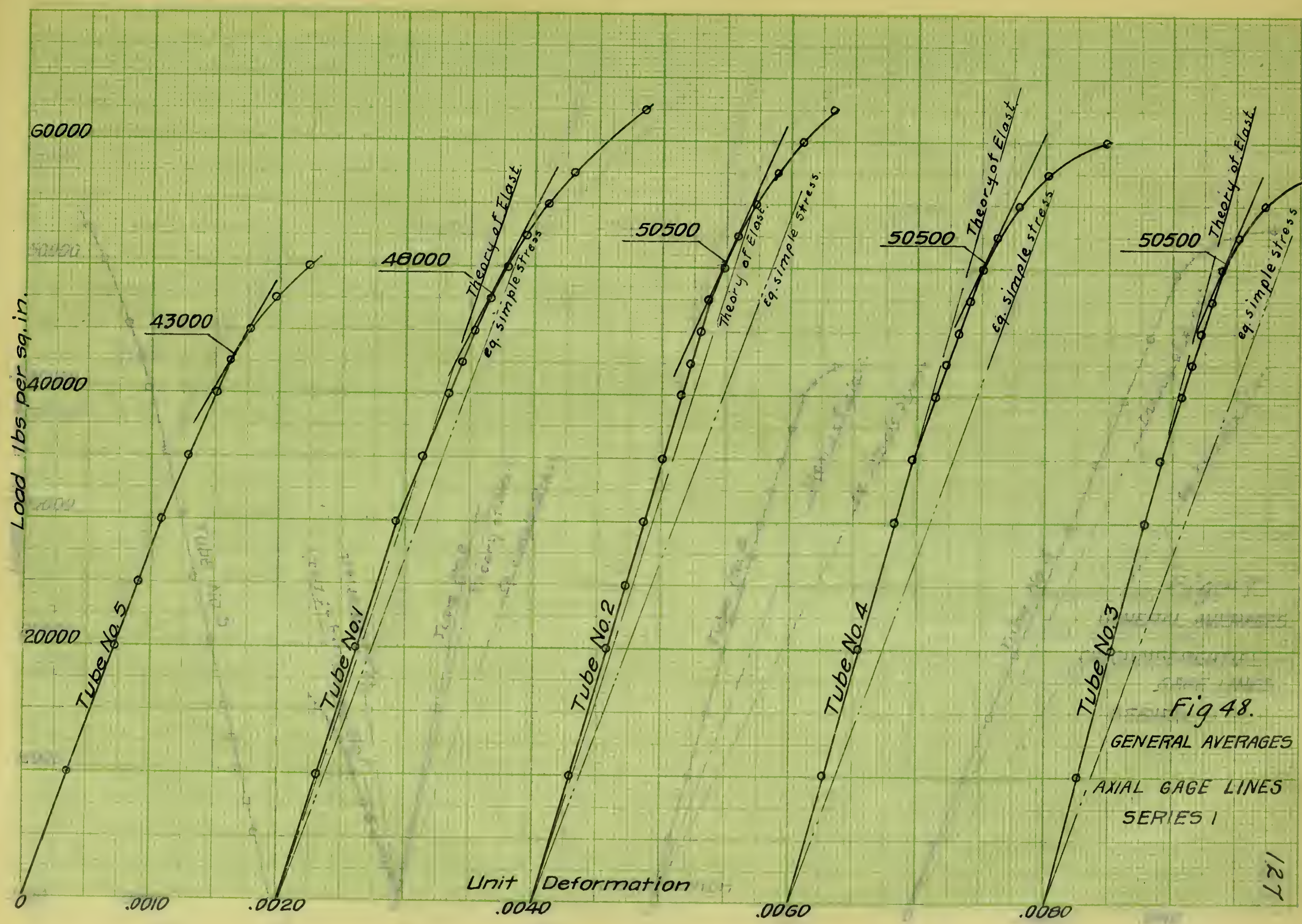
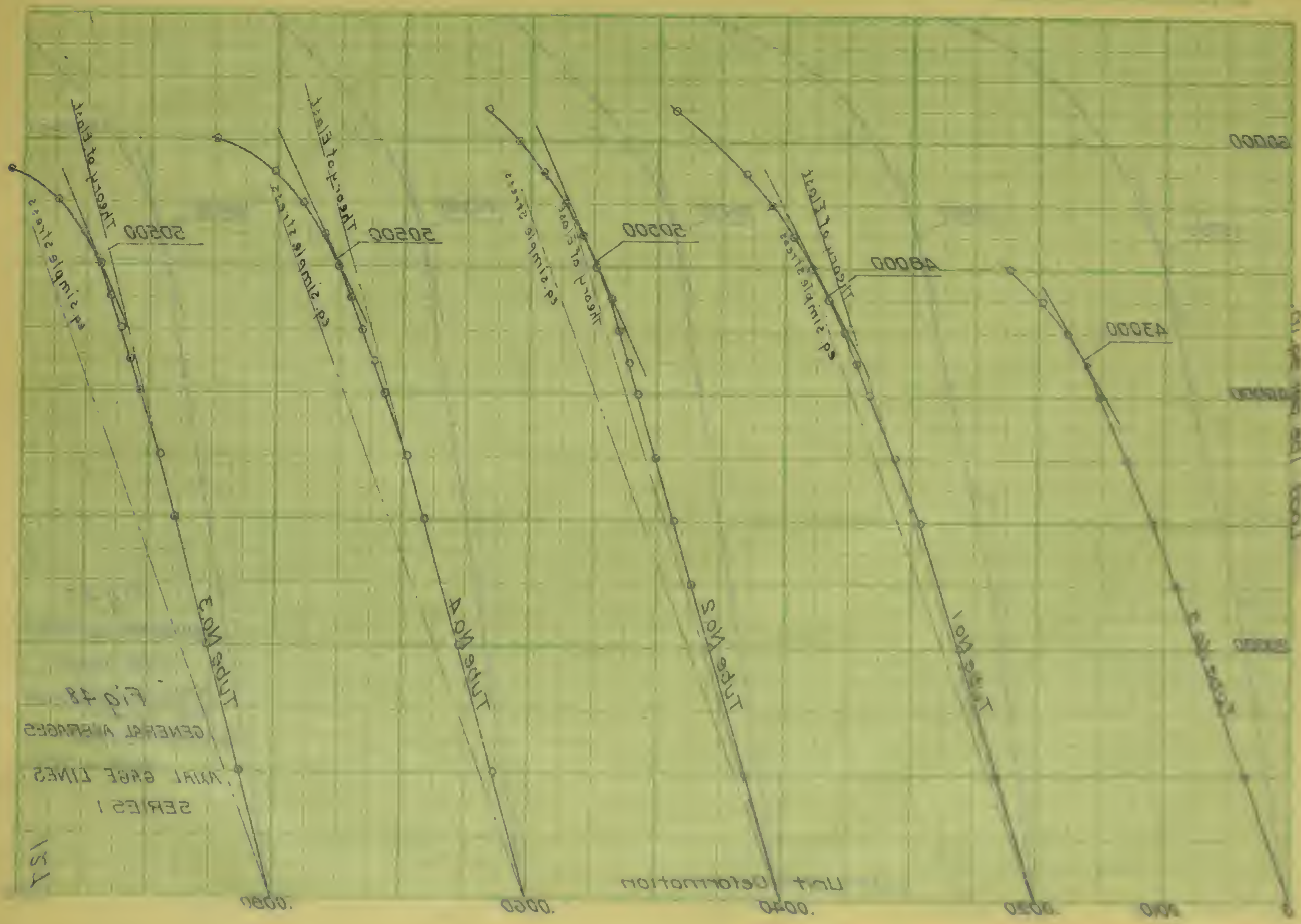


Fig 48.
GENERAL AVERAGES
AXIAL GAGE LINES
SERIES I


127
Fig 43
GENERAL AVERAGES
AXIAL GAGE LINES
SERIES I
Tube No 3
Tube No 4
Tube No 2
Tube No 1
Theory of Elast
eq. simple stress
50500
50500
50500
48000
43000
50000
Unit Deformation
.0040
.0020

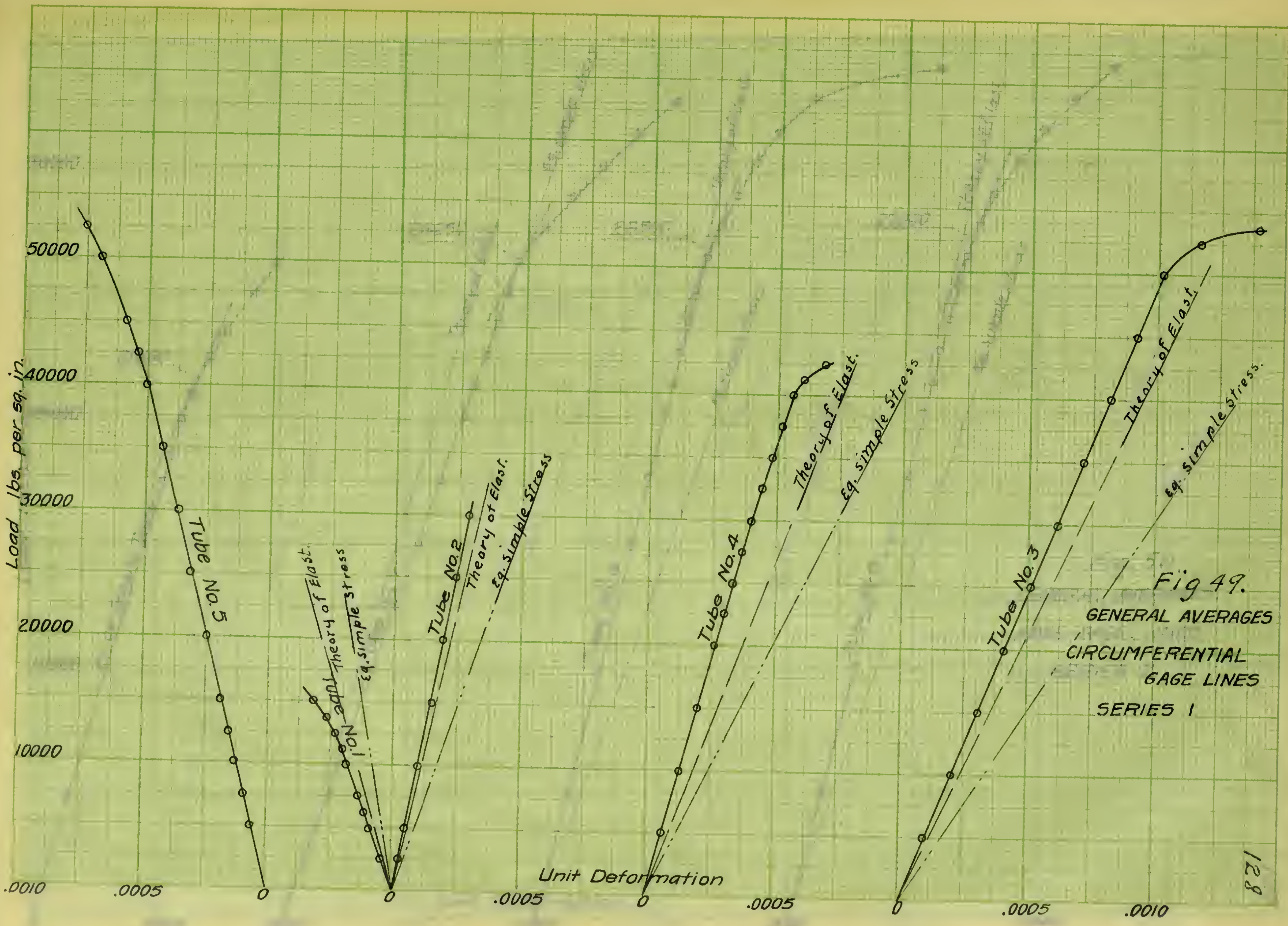


Fig 49.
GENERAL AVERAGES
CIRCUMFERENTIAL GAGE LINES
SERIES 1

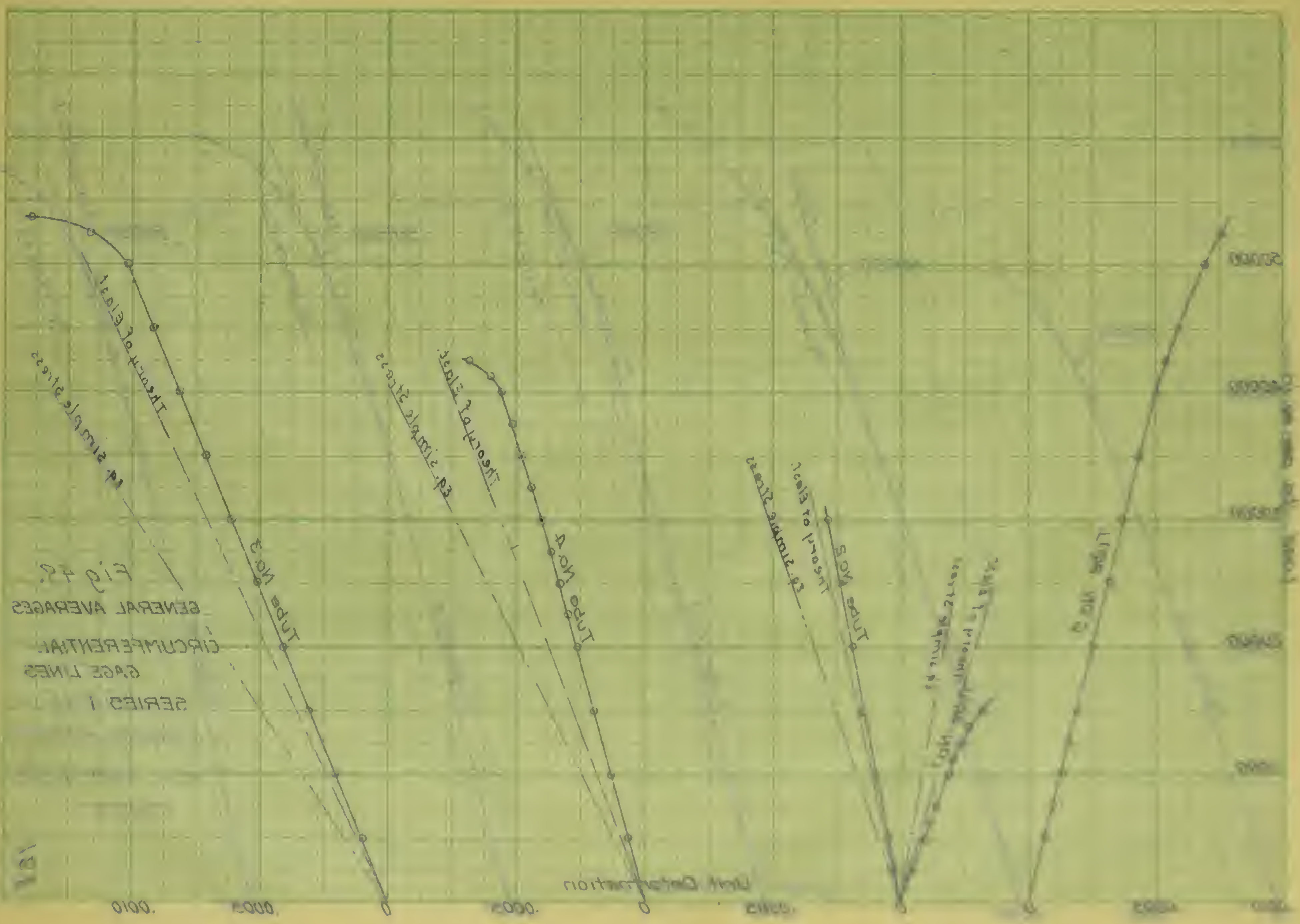


Fig 45.
GENERAL AVERAGES
CIRCUMFERENTIAL
GAGE LINES
SERIES I
Tube No. 3
Tube No. 4
Tube No. 2
Tube No. 5
Theory of Elast.
Eq. simple stress
Unit Deformation
Load lbs. per sq. in.
0
.0005
.0010
5000
10000
15000
20000
25000
30000
181

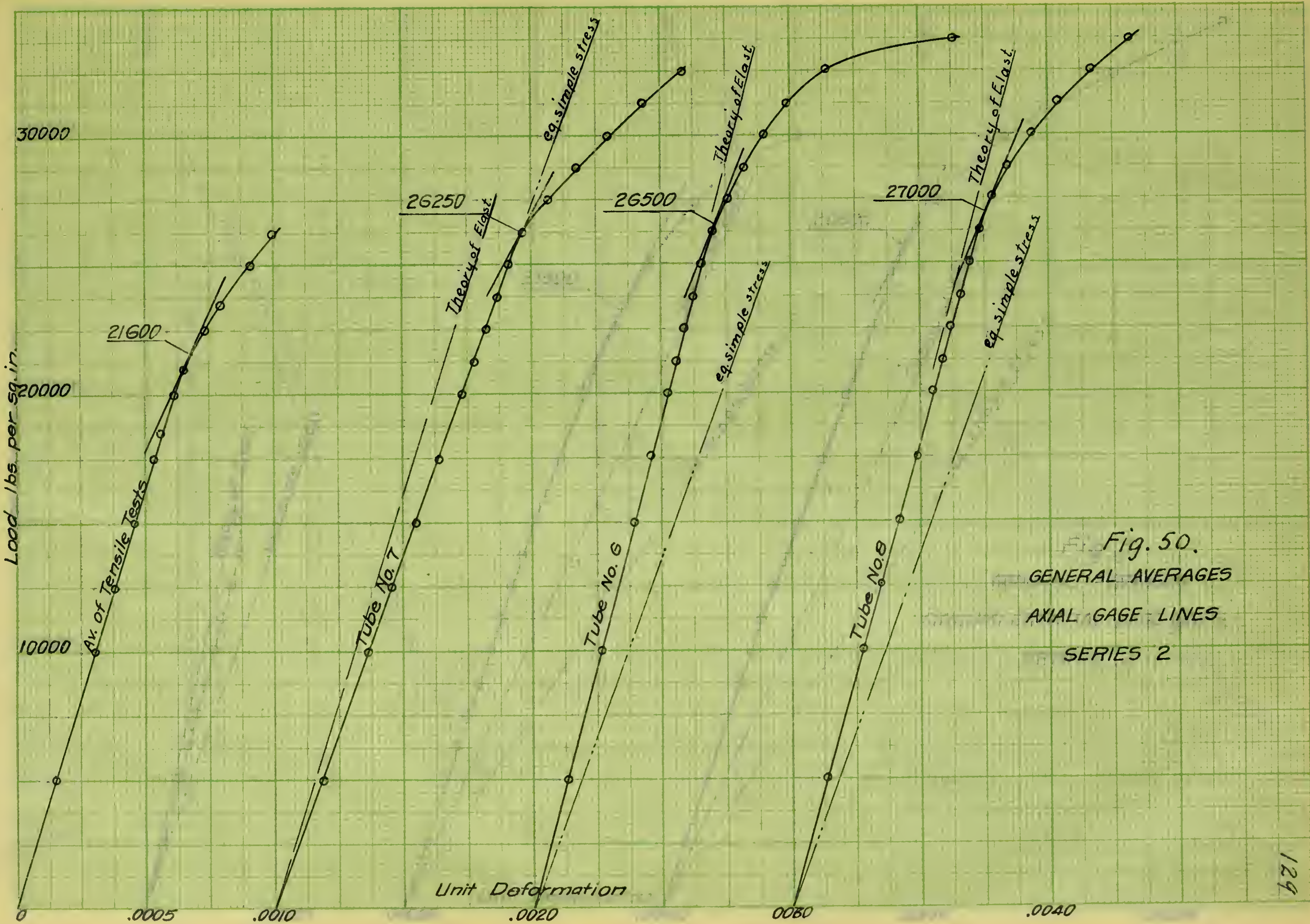


Fig. 50.
GENERAL AVERAGES
AXIAL GAGE LINES
SERIES 2

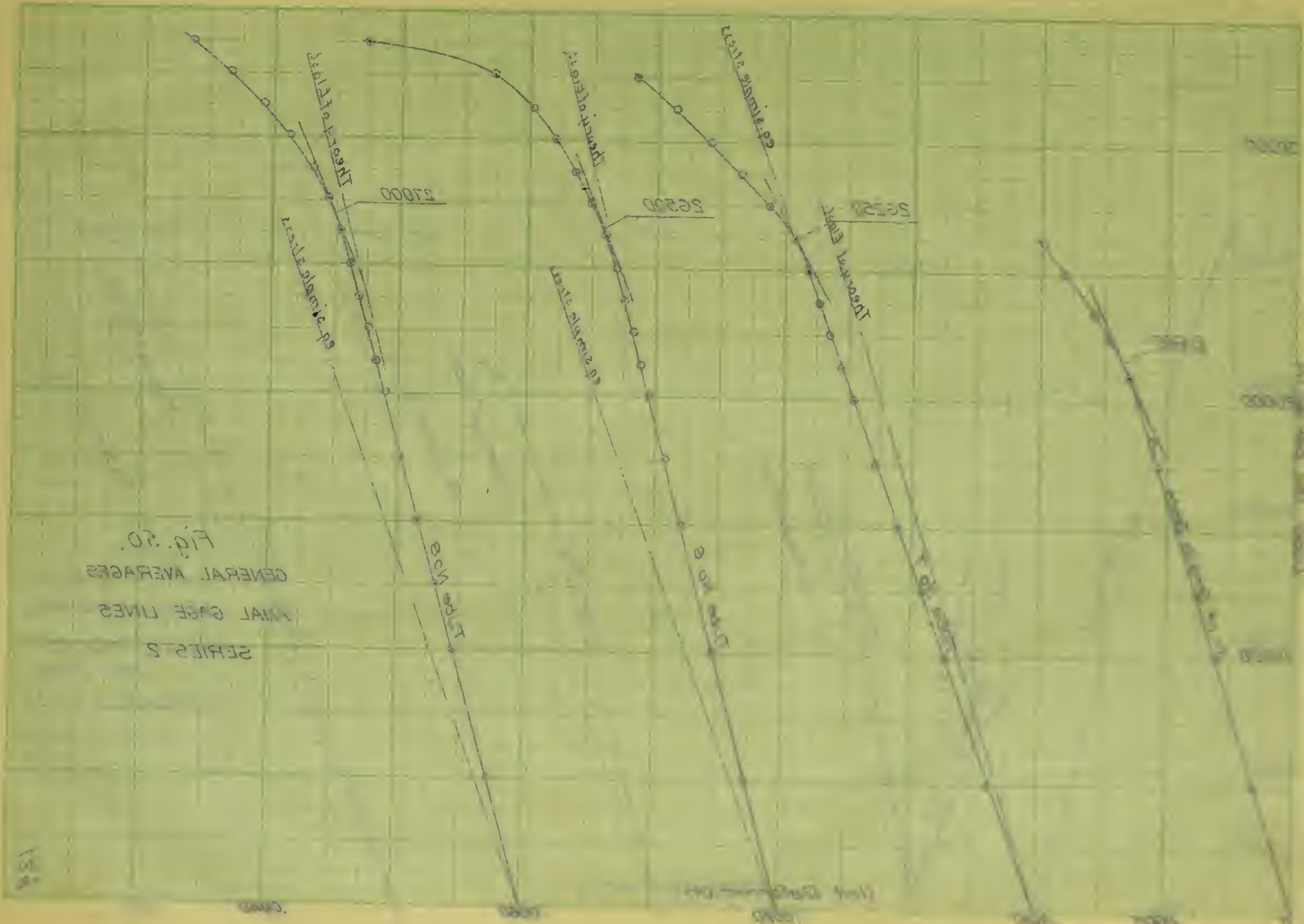


Fig. 50.
GENERAL AVERAGES
JOINT GAGE LINES
SERIES 2

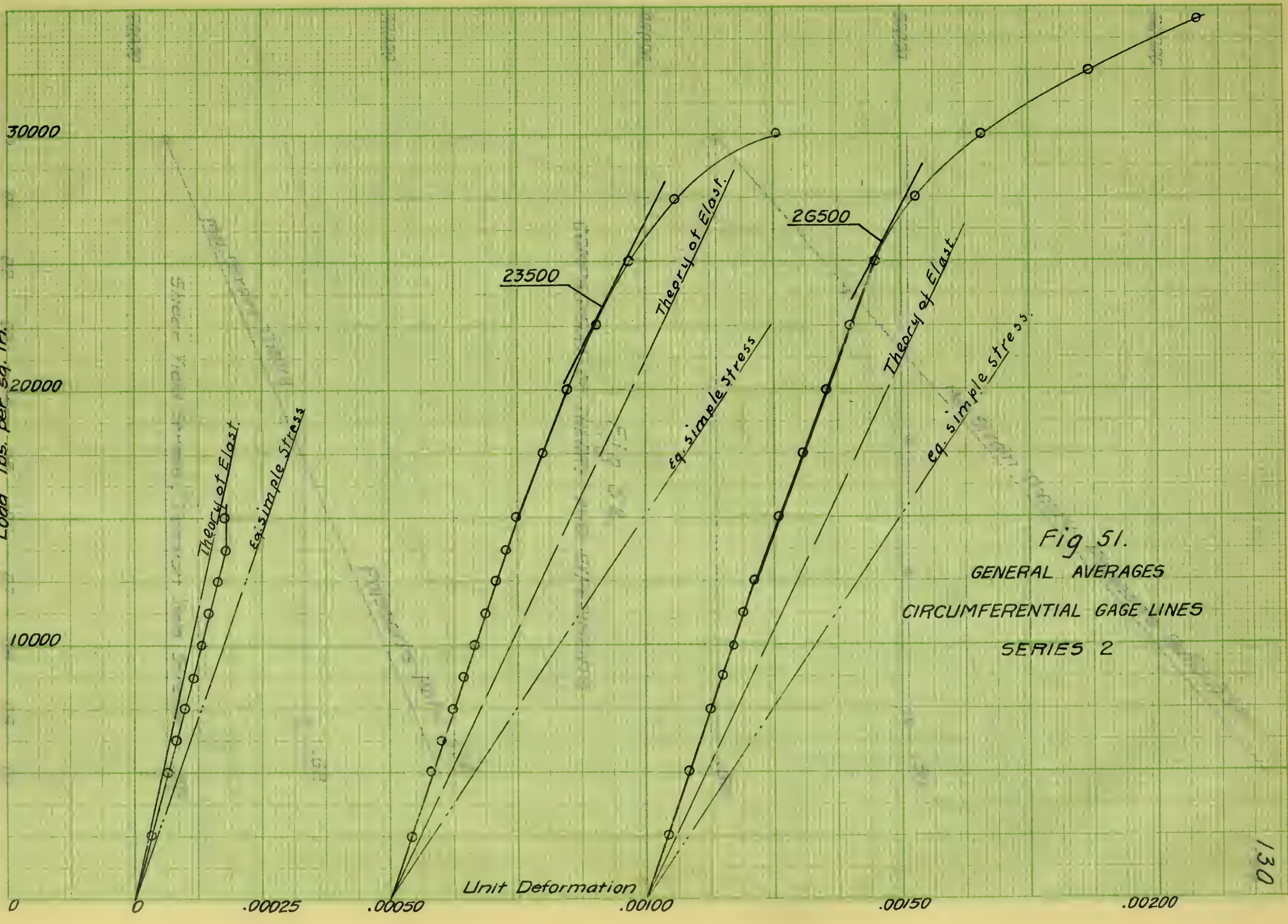


Fig. 51.
GENERAL AVERAGES
CIRCUMFERENTIAL GAGE LINES
SERIES 2

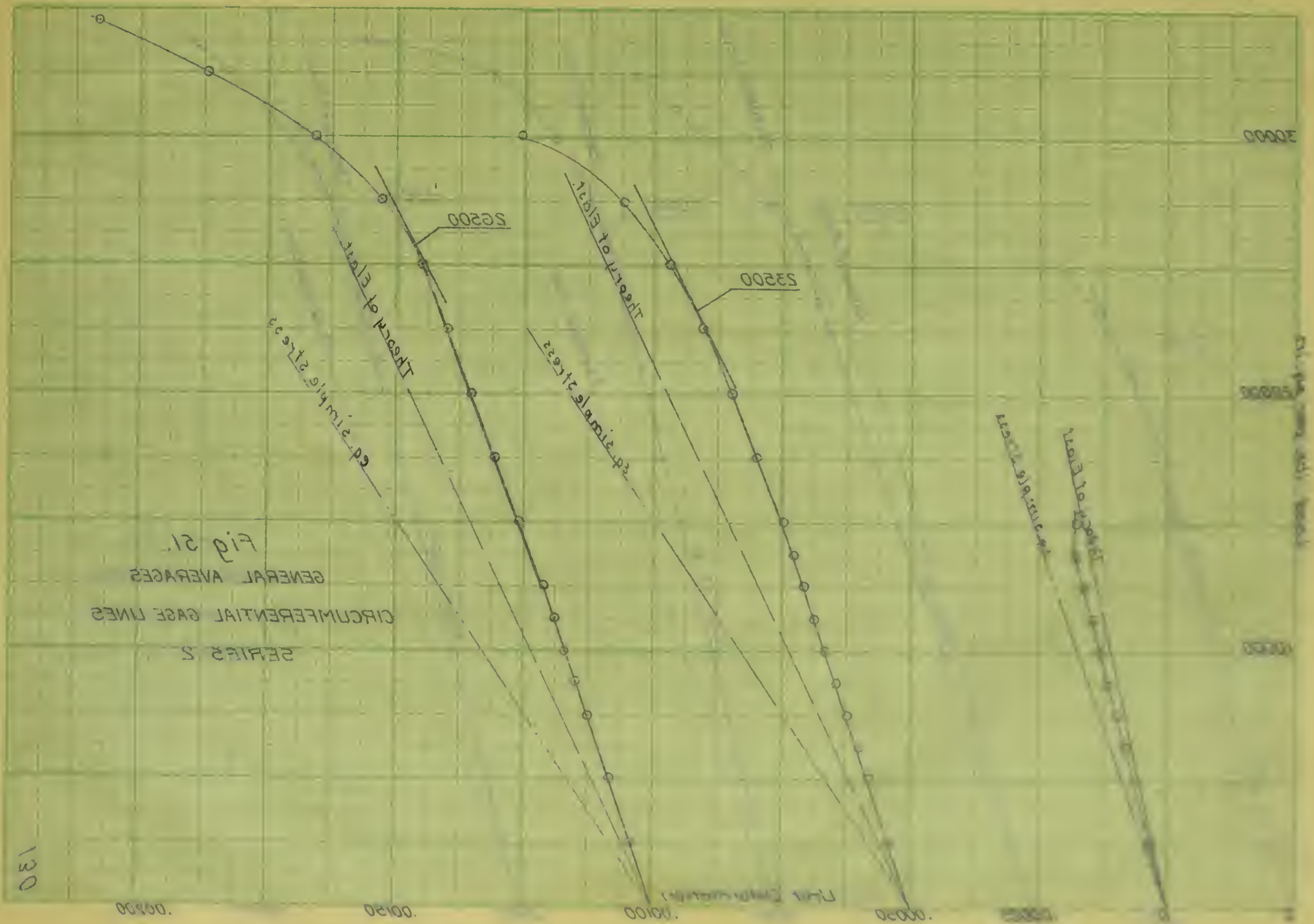


Fig 15.
GENERAL AVERAGES
CIRCUMFERENTIAL GAGE LINES
SERIES 2
26500
23500
Theory of Elast.
Theory of Elast.
Theory of Elast.
eq. simple stress
eq. simple stress
eq. simple stress
30000
20000
10000
.00050
.00100
.00150
.00200
Unit Deformation

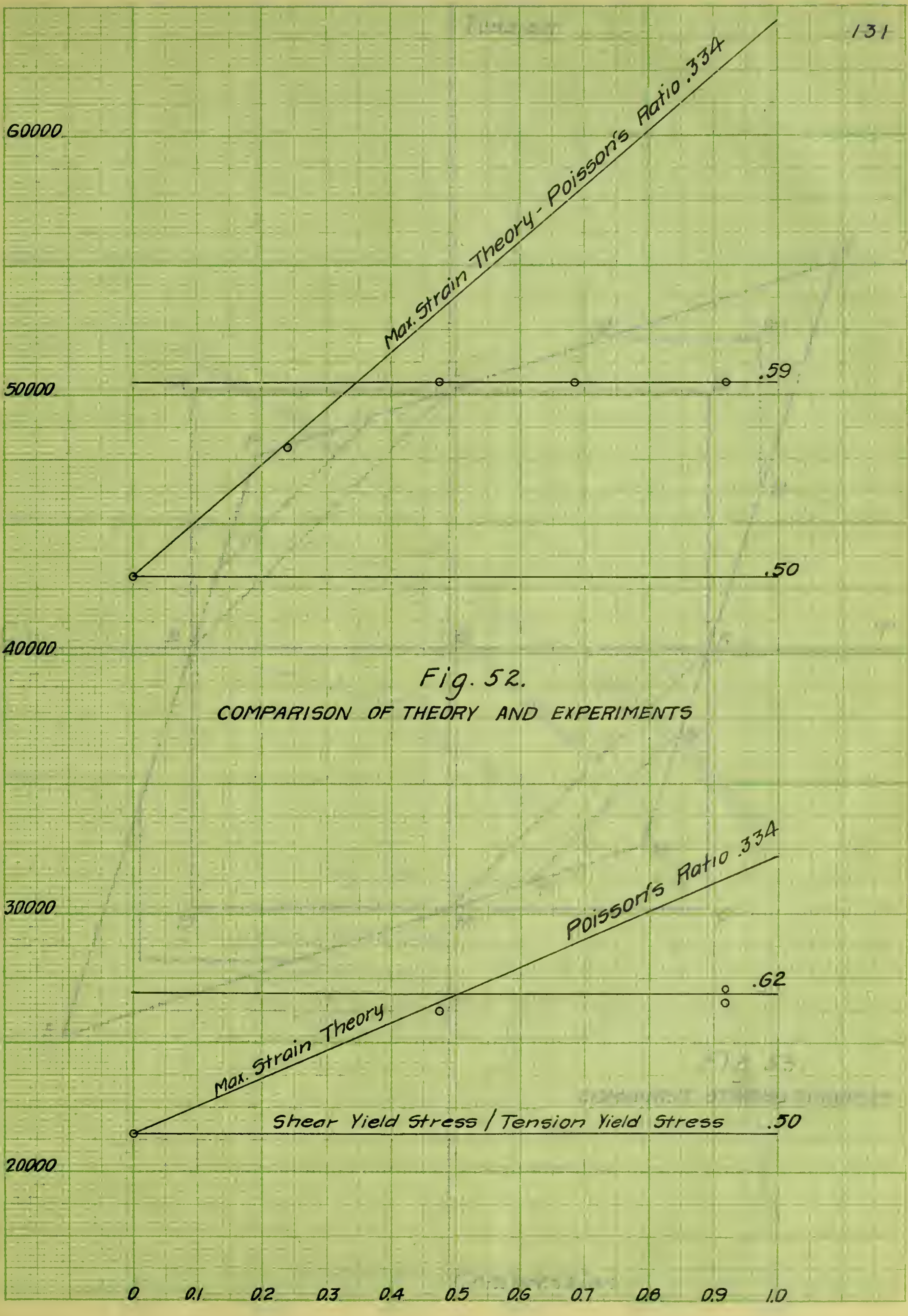


Fig. 52.

COMPARISON OF THEORY AND EXPERIMENTS

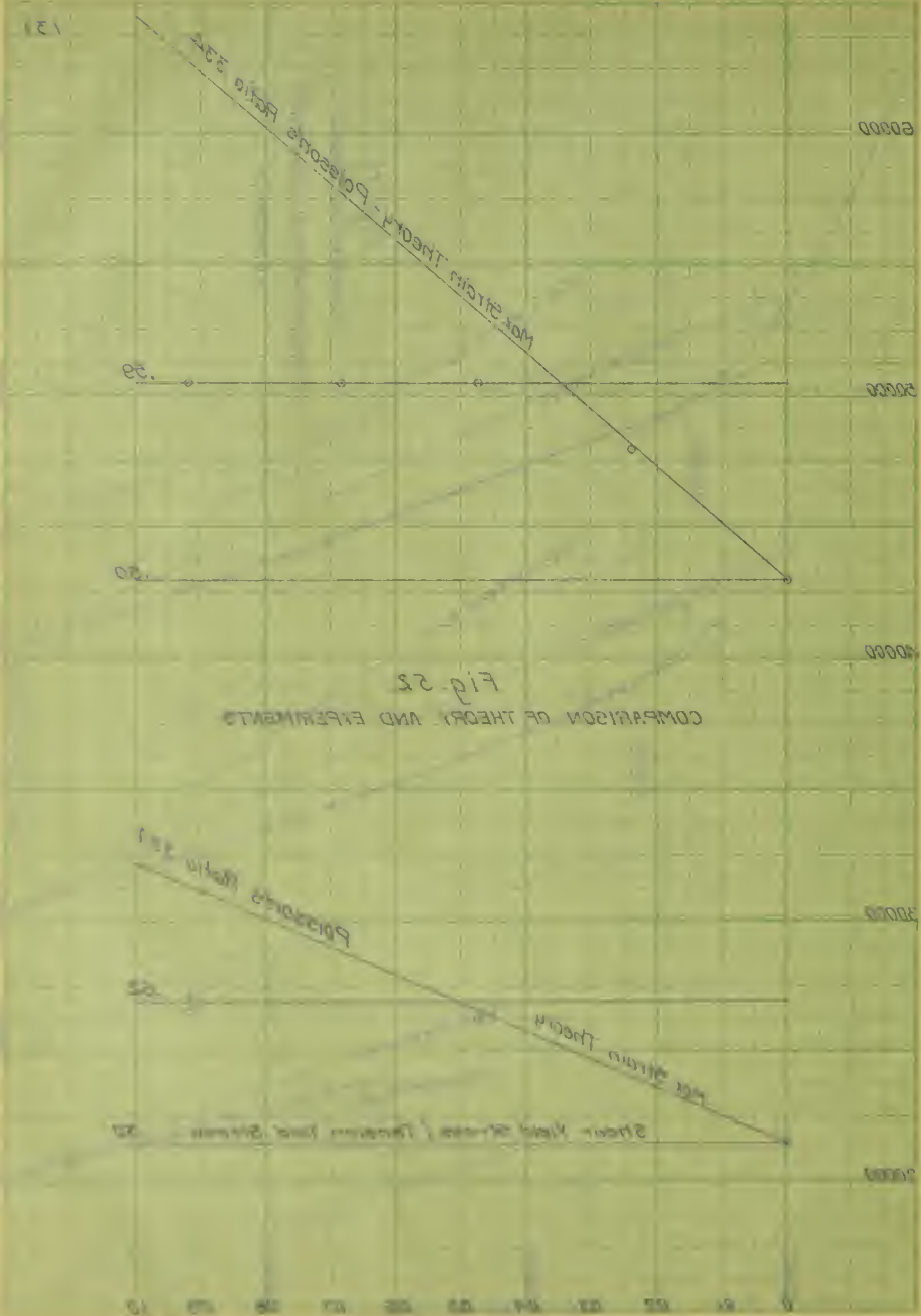


Fig. 52
COMPARISON OF THEORY AND EXPERIMENTS

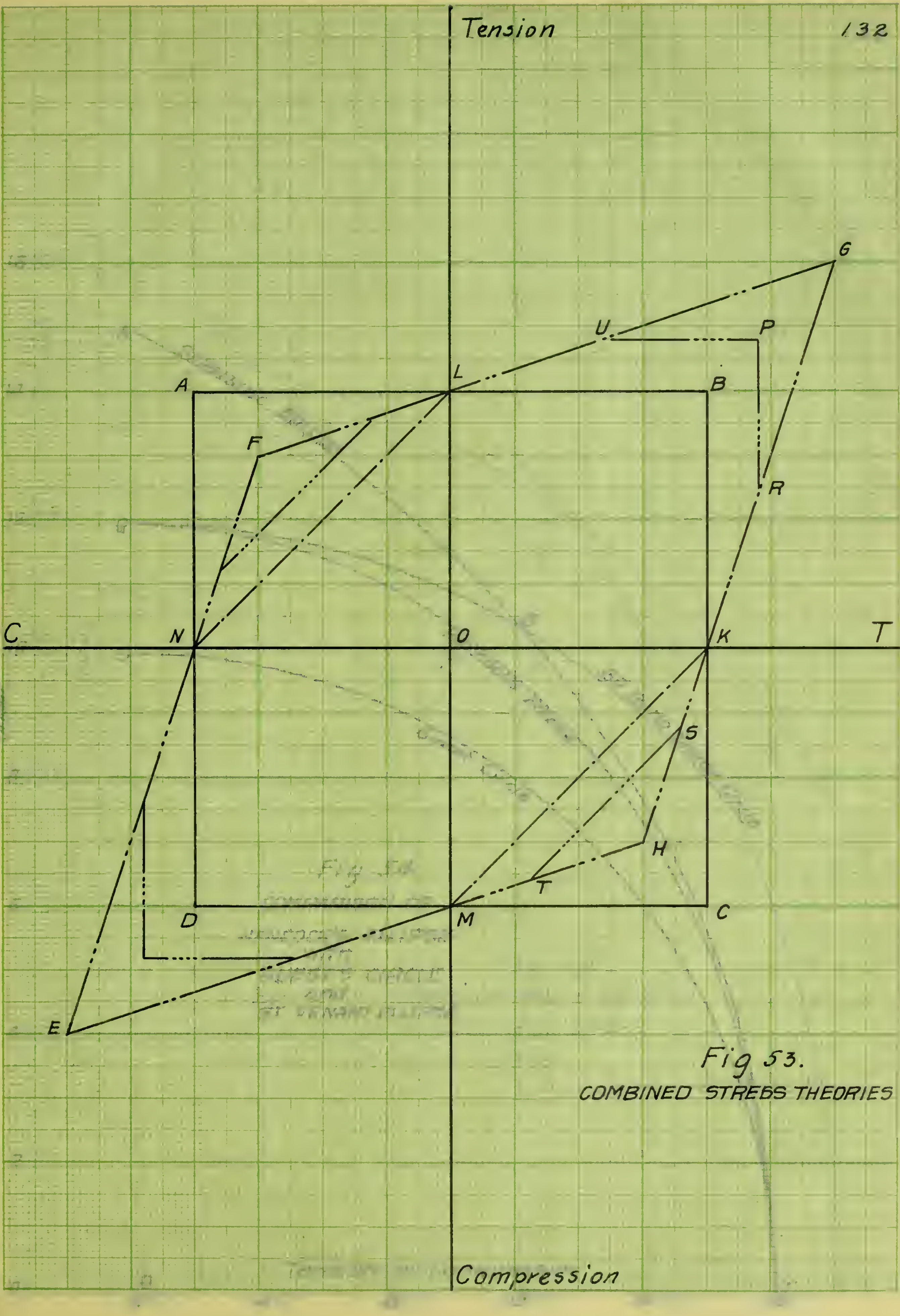


Fig 53.
COMBINED STRESS THEORIES

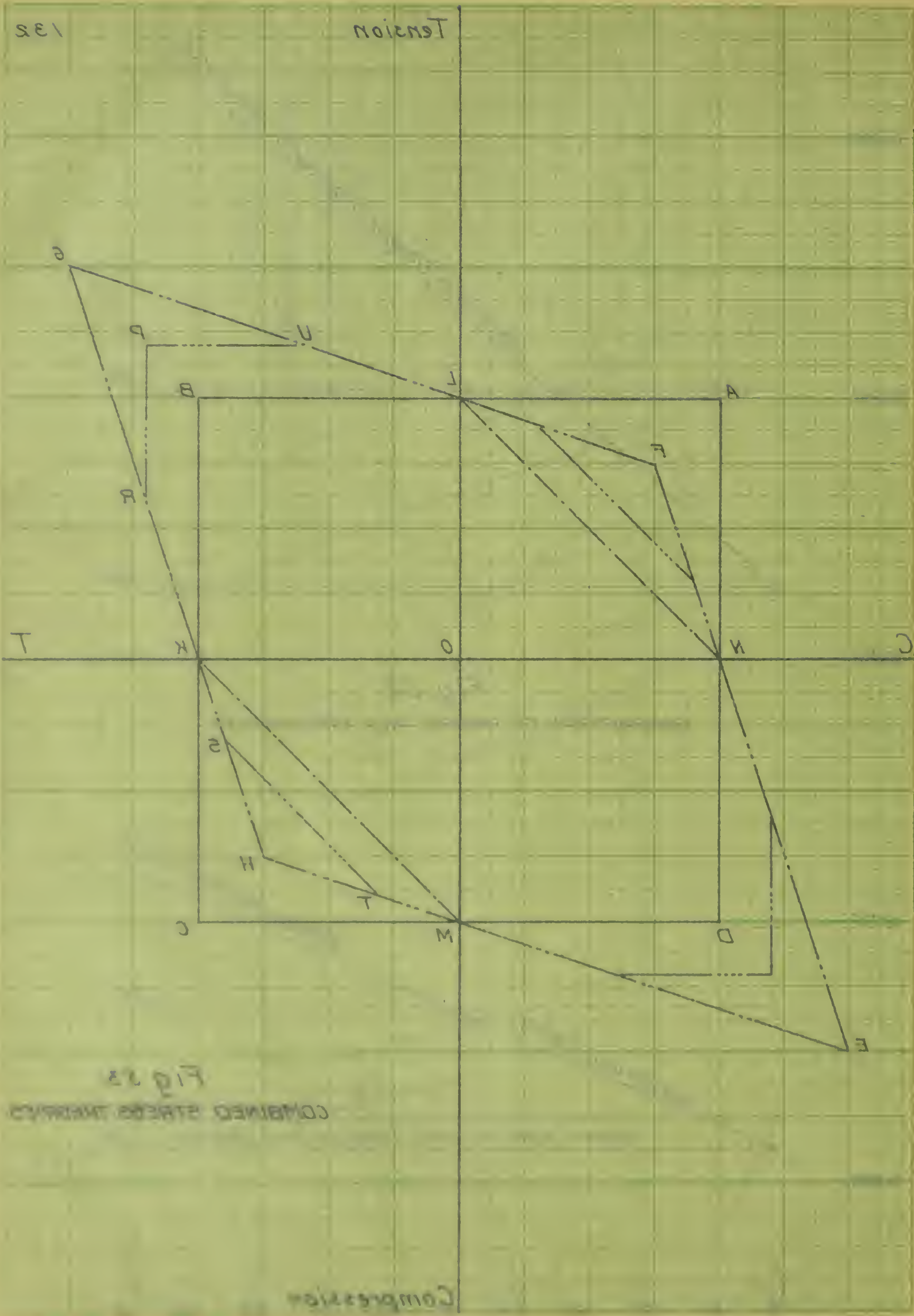


Fig 53
COMBINED STRESS THEORIES

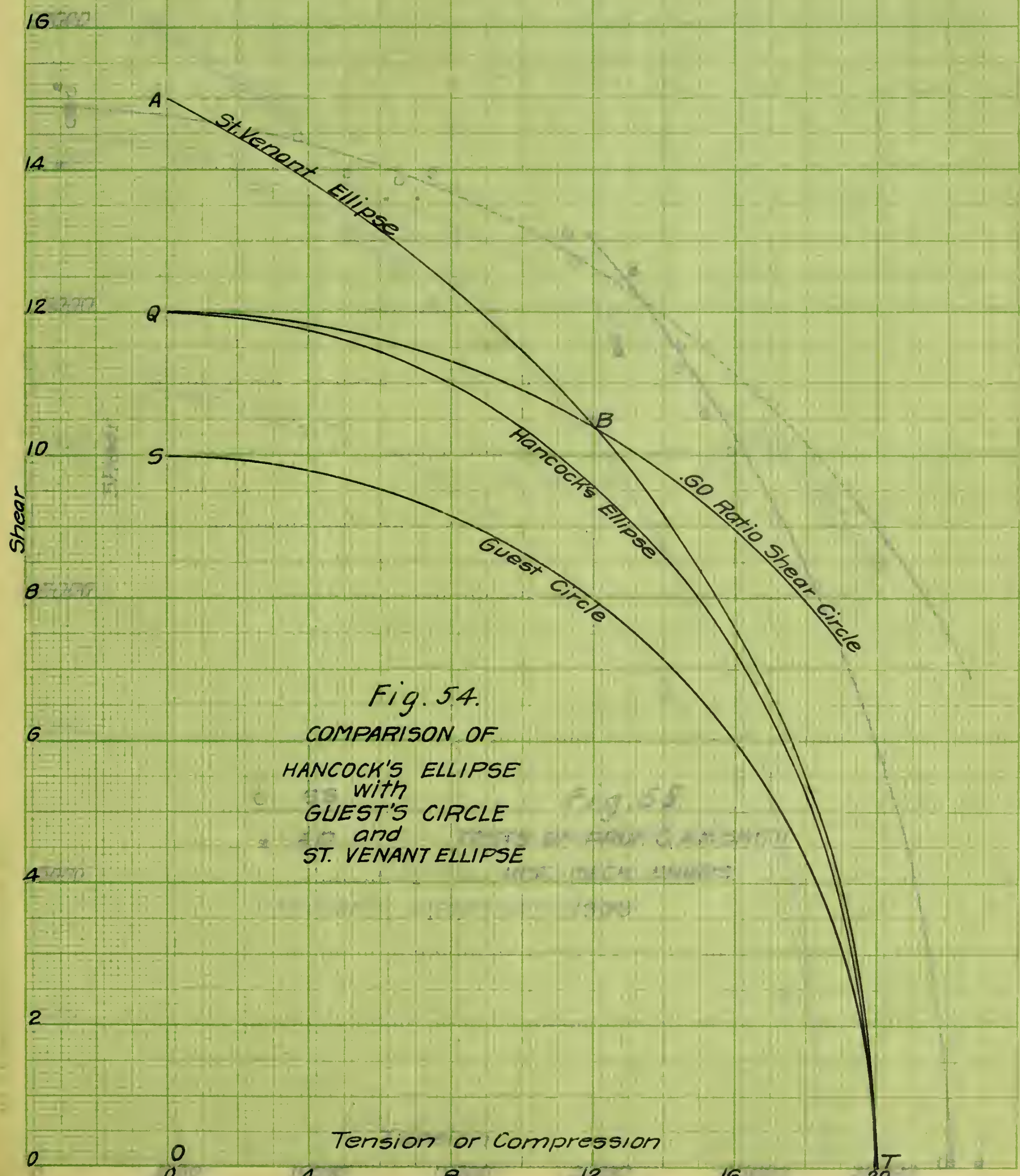


Fig. 54.
COMPARISON OF
HANCOCK'S ELLIPSE
with
GUEST'S CIRCLE
and
ST. VENANT ELLIPSE

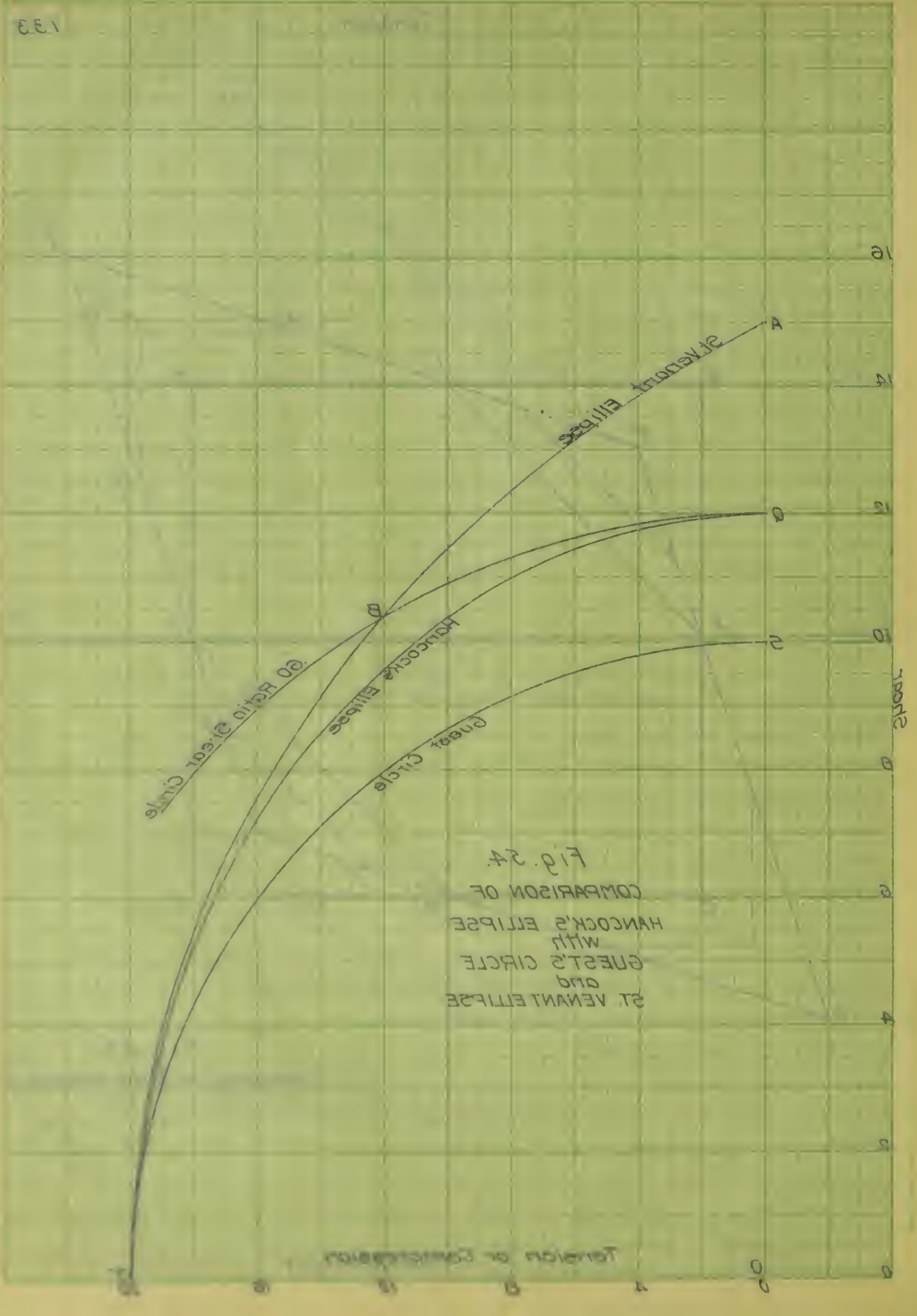


Fig. 54.
COMPARISON OF
HANCOCK'S ELLIPSE
with
GUEST'S CIRCLE
and
ST. VENANT ELLIPSE

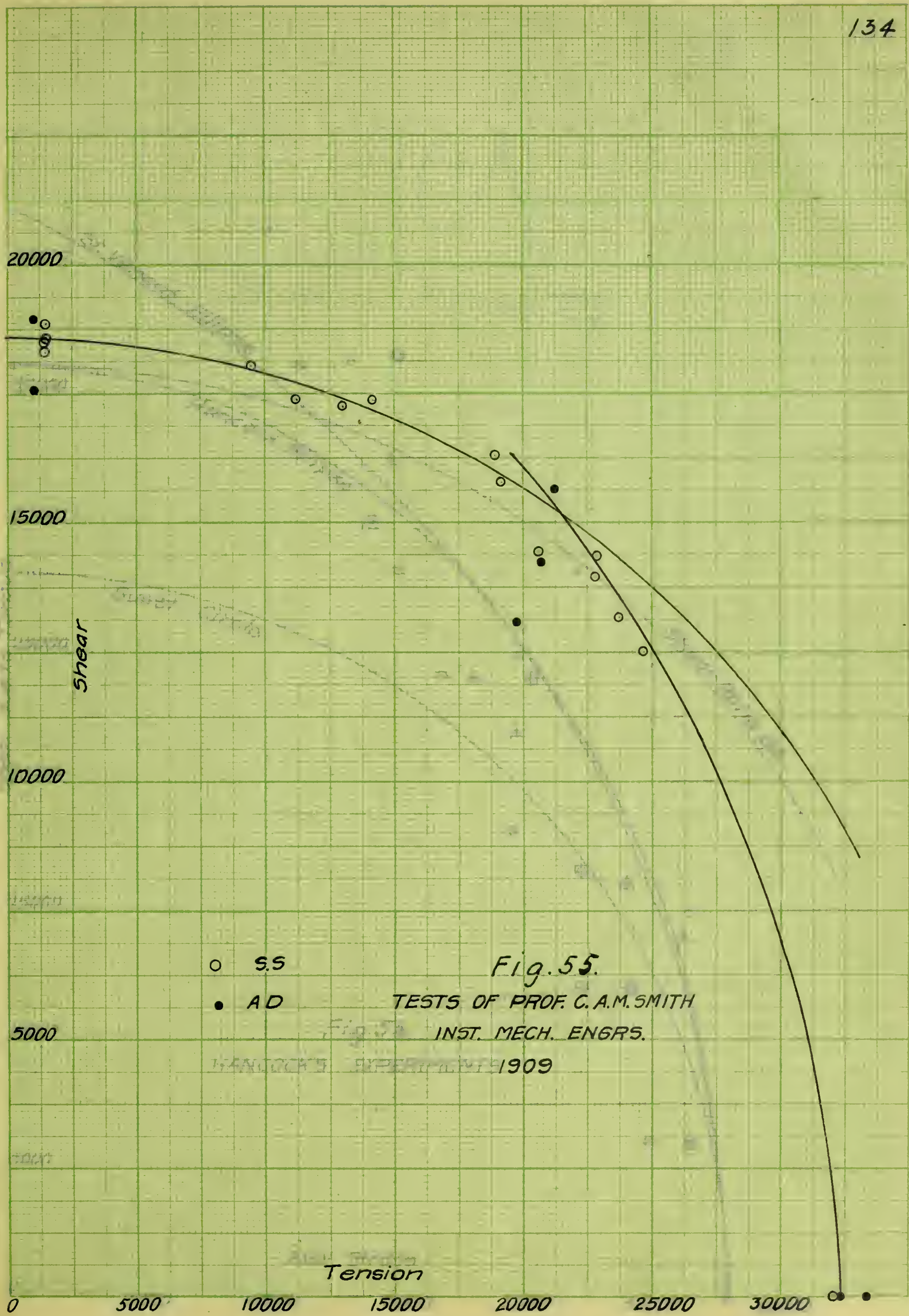


Fig. 55.
TESTS OF PROF. C.A.M. SMITH
INST. MECH. ENGRS.
1909

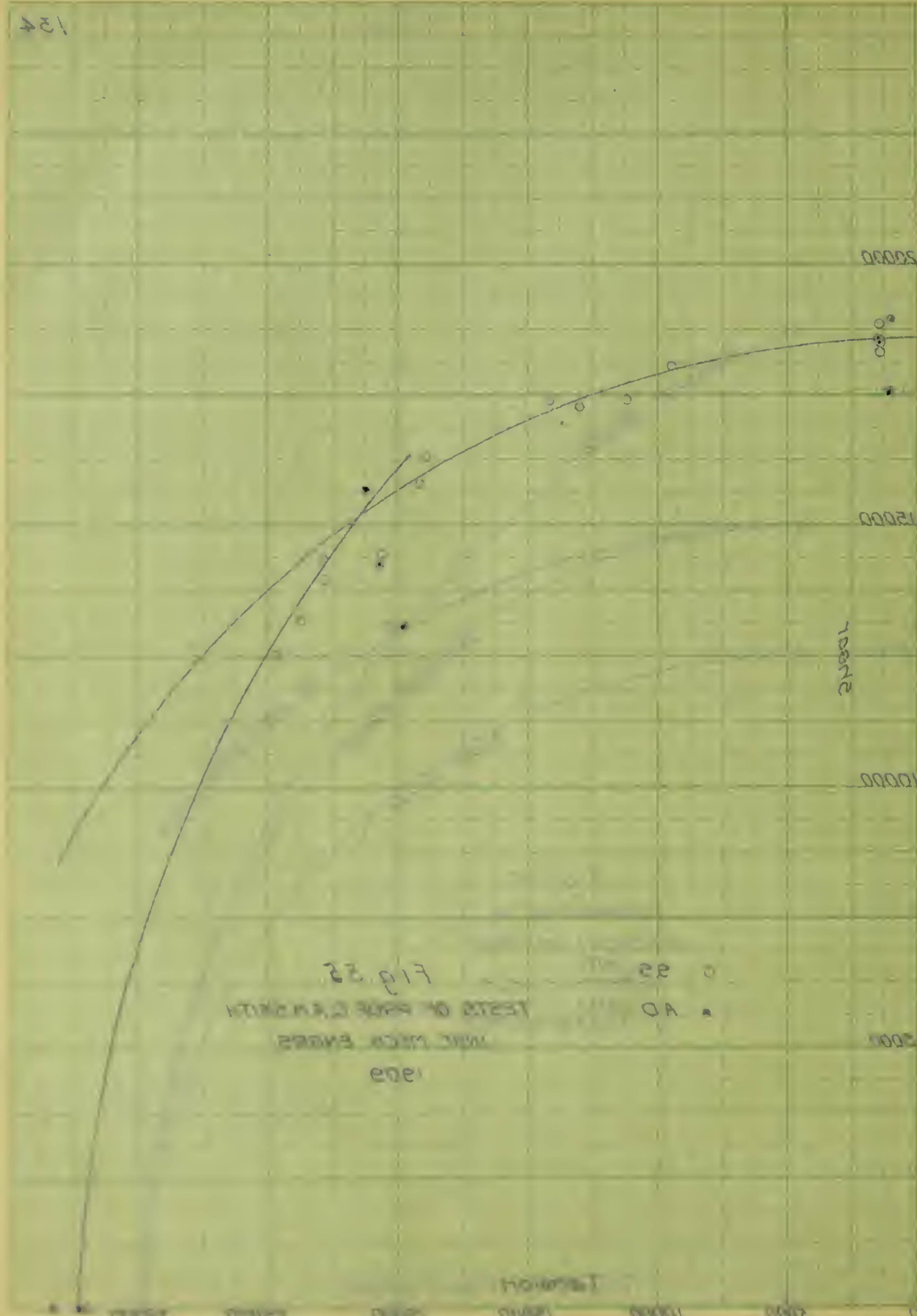


Fig. 55

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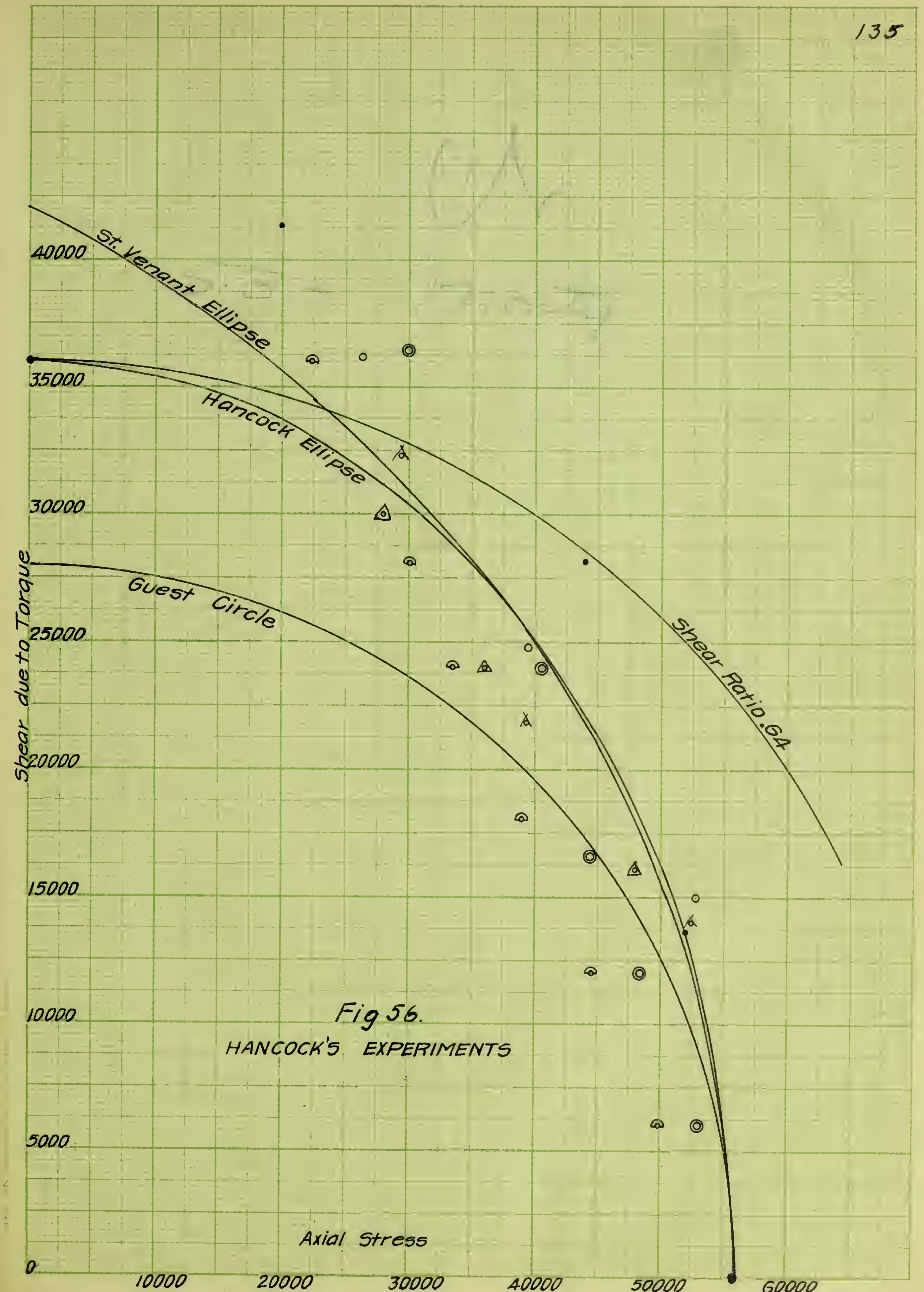

Fig 56.
HANCOCK'S EXPERIMENTS

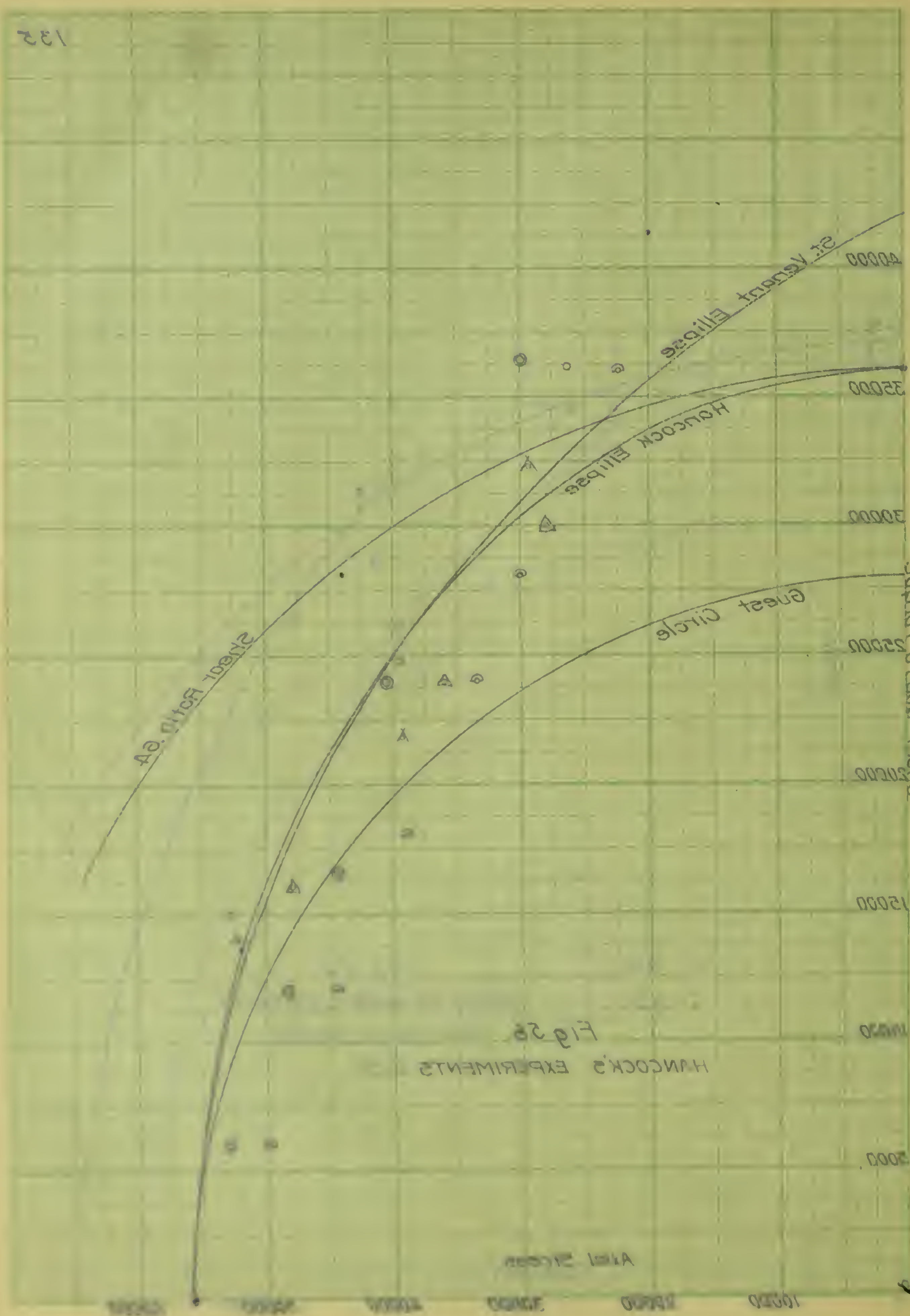

Fig 56
HANCOCK'S EXPERIMENTS

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TABLE I
CROSS BENDING TESTS
First Series
Unit Deformation x 10,000
Arms 3 and 4 Loaded

Gage Lines	Load in Lbs. on Each Arm 32.5	57.5	82.5	107.5	132.5	157.5	182.5	207.5	232.5
13A	1.8	2.1	3.1	3.9	5.0	6.5	7.1	8.6	9.4
35A	.8	1.9	2.9	3.5	4.0	5.1	6.2	7.6	8.3
57A	.9	1.9	3.4	4.0	5.4	7.0	7.6	9.5	10.2
13B	.9	2.0	2.9	4.0	4.9	6.1	6.0	7.5	8.4
24B	.8	1.8	2.5	3.0	3.9	5.0	5.6	7.1	9.0
35B	0	.4	1.4	2.4	3.3	4.2	3.9	5.9	6.8
46B	1.0	2.1	2.7	3.6	4.5	5.3	5.3	7.3	7.9
57B	1.4	1.7	2.5	3.6	4.5	5.7	5.6	7.9	8.8
13C	1.3	2.5	3.5	4.3	5.2	6.8	7.4	9.6	10.7
35C	.4	1.3	2.0	2.9	3.2	5.0	5.3	6.4	6.8
57C	.7	1.7	2.9	3.9	5.1	6.3	6.7	9.5	9.6
13D	-.1	-.2	-.7	-.9	-1.2	-.3	-.1	-1.2	-1.4
35D	-.3	-.6	-.5	-1.6	-1.0	-1.5	-.8	-1.1	-1.7
57D	-.6	-.4	-.7	-1.1	-1.1	-1.4	-.4	-1.3	-1.5
13E	-.1	0	-.1	1.0	-.8	-.9	-.4	-1.2	-1.4
24E	-.1	-.3	-.2	-.7	-.5	-.6	-.7	-1.3	-1.3
35E	-.1	-.2	-.2	-.9	-.9	-1.0	-.5	-1.0	-.9
46E	.2	-.1	-.3	-.3	-.6	-.6	-.4	-.7	-.8
57E	-.2	-.6	-.7	-1.2	-1.0	-1.4	-.6	-.5	-1.1
13K	-.6	-1.0	-1.3	-1.7	-1.9	-2.1	-.8	-2.3	-2.6
35K	-.3	-.4	-.5	-1.2	-1.3	-1.4	-.7	-1.1	-1.2
57K	-.9	-1.3	-1.5	-1.8	-1.2	-2.1	-1.6	-1.8	-1.9
34H	.4	.7	1.0	.9	1.7	1.6	2.7	2.9	3.0
45H	.4	.3	1.0	1.0	1.8	1.9	2.8	3.1	3.1
34N	.6	.5	.7	.5	1.0	1.3	1.5	1.8	1.8
45N	0	.4	.5	.4	.9	.8	1.3	1.2	1.2

TABLE II
CROSS BENDING TESTS
First Series
Unit Deformation x 10,000
All Four Arms Loaded

Gage Lines	Load in Lbs. on Each Arm 32.5	57.5	82.5	107.5	132.5	157.5	182.5
13A	.7	1.2	2.1	3.1	4.6	5.7	7.8
35A	.8	.9	1.3	1.8	2.7	4.0	4.0
57A	1.4	2.0	2.9	3.8	4.8	6.6	7.4
13B	1.4	1.7	2.7	3.7	4.6	5.9	6.6
24B	.8	1.2	2.0	3.1	3.7	5.1	5.7
35B	.7	1.0	1.9	2.5	3.0	4.6	4.8
46B	1.0	1.4	2.0	2.8	3.1	4.3	4.8
57B	1.1	1.3	2.1	3.2	4.1	5.1	5.9
13C	1.1	1.3	2.1	3.1	4.2	5.2	6.1
35C	.5	.8	1.5	1.7	2.6	3.5	3.4
57C	.8	1.9	2.8	3.9	4.9	5.8	6.7
13D	.7	1.3	1.9	3.0	3.8	4.7	5.2
35D	.4	.4	.6	1.5	1.8	2.6	3.2
57D	1.1	2.0	3.0	4.1	4.9	6.2	7.0
13E	.8	1.5	1.9	3.6	4.3	5.3	5.4
24E	.9	1.3	1.8	2.4	3.1	3.9	4.1
35E	.9	1.1	1.2	2.2	3.1	3.7	4.0
46E	.9	1.4	1.9	3.0	4.0	4.9	5.7
57E	1.2	2.4	3.3	4.5	5.5	7.4	5.3
13K	.8	1.4	2.0	3.2	4.3	6.5	6.0
35K	.3	.6	1.5	2.6	3.0	4.2	3.8
57K	1.1	2.1	2.7	4.0	4.8	5.9	6.9
34H	.6	1.6	2.0	2.6	3.3	3.2	3.9
45H	.2	.7	1.4	2.1	2.3	2.7	3.2
34N	.5	1.0	1.1	1.9	2.0	3.7	3.6
45N	.6	1.1	1.5	1.8	1.8	3.2	3.3

TABLE II
CROSS BENDING TESTS
(Continued)

Gage Lines	Load in Lbs. on Each Arm 207.5	265	290	315	340	365	375
13A	8.0	9.9	11.3	12.5	14.6	20.0	41.8
35A	4.1	5.6	6.0	6.2	7.2	8.2	12.2
57A	8.6	10.5	11.6	12.9	14.4	17.1	25.0
13B	7.7	10.0	11.0	12.2	13.5	20.4	42.0
24B	6.5	8.4	9.4	10.2	12.1	17.9	38.8
35B	5.9	6.8	8.4	9.1	10.4	11.7	13.7
46B	5.9	7.3	7.7	8.6	10.0	12.0	19.7
57B	7.1	8.9	9.5	10.6	11.7	15.1	20.0
13C	7.1	10.9	12.3	13.9	15.9	23.5	44.1
35C	4.4	5.3	5.2	5.4	6.1	6.9	11.2
57C	7.7	9.5	10.1	11.3	13.0	15.5	21.4
13D	6.6	8.4	9.3	10.4	14.4	15.3	17.5
35D	3.5	4.3	4.7	5.0	5.7	7.1	11.3
57D	8.3	10.8	11.4	13.2	15.7	20.1	34.4
13E	6.5	8.4	9.4	10.5	12.4	18.8	18.3
24E	4.9	6.6	6.7	8.2	10.5	11.7	17.4
35E	5.0	6.8	6.6	7.7	10.2	12.8	15.1
46E	6.5	8.6	8.8	10.0	12.7	16.3	34.4
57E	7.1	8.2	9.4	9.8	11.9	16.1	29.6
13K	6.9	9.8	10.4	12.0	15.2	18.2	23.2
35K	5.3	5.2	4.9	5.4	7.6	7.5	8.9
57K	7.8	10.8	11.5	12.9	15.8	22.0	35.7
34H	4.4	5.0	4.7	4.8	6.7	8.0	11.8
45H	3.6	4.8	3.7	4.6	5.0	5.1	7.8
34N	4.8	5.9	5.5	6.5	7.3	8.9	12.6
45N	4.2	6.3	6.0	7.0	7.7	8.5	12.0

TABLE III
COMPARATIVE DEFORMATIONS
With 2 and 4 Arms Loaded
First Series
Deformations x 10,000

Gage Lines	Load in Lbs. $32\frac{1}{2}$	$57\frac{1}{2}$	$82\frac{1}{2}$	107.5	132.5	157.5	182.5	207.5	Load on Arms
13A	1.8	2.1	3.1	3.9	5.0	6.5	7.1	8.6	2
13A	.7	1.2	2.1	3.1	4.6	5.7	7.8	8.0	4
35A	.8	1.9	2.9	3.5	4.0	5.1	6.2	7.6	2
35A	.8	..9	1.3	1.8	2.7	4.0	4.0	4.1	4
57A	.9	1.9	3.4	4.0	5.4	7.0	7.6	9.5	2
57A	1.4	2.0	2.9	3.8	4.8	6.6	7.4	8.6	4
13B	.9	2.0	2.9	4.0	4.9	6.1	6.0	7.5	2
13B	1.4	1.7	2.7	3.7	4.6	5.9	6.6	7.7	4
24B	.8	1.8	2.5	3.0	3.9	5.0	5.6	7.1	2
24B	.8	1.2	2.0	3.1	3.7	5.1	5.7	6.5	4
35B	0	.4	1.4	2.4	3.3	4.2	3.9	5.9	2
35B	.7	1.0	1.9	2.5	3.0	4.6	4.8	5.9	4
46B	1.0	2.1	2.7	3.6	4.5	5.3	5.3	7.3	2
46B	1.0	1.4	2.0	2.8	3.1	4.3	4.8	5.9	4
57B	1.4	1.7	2.5	3.6	4.5	5.7	5.6	7.9	2
57B	1.1	1.3	2.1	3.2	4.1	5.1	5.9	7.1	4
13C	1.3	2.5	3.5	4.3	5.2	6.8	7.4	9.6	2
13C	1.1	1.3	2.1	3.1	4.2	5.2	6.1	7.1	4
35C	.4	1.3	2.0	2.9	3.2	5.0	5.3	6.4	2
35C	.5	.8	1.5	1.7	2.6	3.5	3.4	4.4	4
57C	.7	1.7	2.9	3.9	5.1	6.3	6.7	9.5	2
57C	.8	1.9	2.8	3.9	4.9	5.8	6.7	7.7	4

TABLE IV
First Series
Poisson's Ratio
Deformation x 10,000

Load in Lbs.

Gage Lines	32.5	57.5	82.5	107.5	132.5	157.5	182.5	207.5	232.5
24B	.8	1.8	2.5	3.0	3.9	5.0	5.6	7.1	9.0
0.3x24B	.24	.54	.75	.9	1.2	1.5	1.7	2.1	2.7
35D	.3	.6	.5	1.6	1.0	1.5	.8	1.1	1.7
35B	0	.4	1.4	2.4	3.3	4.2	3.9	5.9	6.8
0.3x35B	0	.12	.4	.7	1.0	1.3	1.2	1.8	2.0
35E	.1	.2	.2	.9	.9	1.0	.5	1.0	.9
46B	1.0	2.1	2.7	3.6	4.5	5.3	5.3	7.3	7.9
0.3x46B	.3	.6	.8	1.1	1.3	1.6	1.6	2.2	2.4
35K	.3	.4	.5	1.2	1.3	1.4	.7	1.1	1.2

TABLE V
Unit Deformations Along and
Across Line B x 10,000

Loads in Lbs.	Gage Lines 13B	24B	35D	35B	35E	46B	35K	57B
32.5	1.4	.8	.4	.7	.9	1.0	.3	1.1
57.5	1.7	1.2	.4	1.0	1.1	1.4	.6	1.3
82.5	2.7	2.0	.6	1.9	1.2	2.0	1.5	2.1
107.5	3.7	3.1	1.5	2.5	2.2	2.8	2.6	3.2
132.5	4.6	3.7	1.8	3.0	3.1	3.1	3.0	4.1
157.5	5.9	5.1	2.6	4.6	3.7	4.3	4.2	5.1
182.5	6.6	5.7	3.2	4.8	4.0	4.8	3.8	5.9
207.5	7.7	6.5	3.5	5.9	5.0	5.9	5.3	7.1
265	10.0	8.4	4.3	6.8	6.8	7.3	5.2	8.9
290	11.0	9.4	4.7	8.4	6.6	7.7	4.9	9.5
315	12.2	10.2	5.0	9.1	7.7	8.6	5.4	10.6
340	13.5	12.1	5.7	10.4	10.2	10.0	7.6	11.7
365	20.4	17.9	7.1	11.7	12.8	12.0	7.5	15.1
375	42.0	38.8	11.3	13.7	15.1	19.7	8.9	20.0

TABLE VI
CROSS BENDING TESTS
Second Series
Unit Deformations x 10,000

Gage Lines	Load in Lbs. on Each Arm 90	165	190	215	265	290	305	320
DH1	.5	1.1	.4	1.0	1.8	2.0	2.0	3.0
FL1	1.0	2.0	0	1.6	2.4	2.3	2.9	2.9
EK3	1.0	2.5	2.1	3.0	4.1	3.3	4.1	5.0
AE5	4.3	7.6	8.3	10.1	13.1	14.9	15.8	17.6
CG5	3.2	5.3	4.8	6.4	8.5	9.1	10.2	11.3
EK5	2.0	3.2	3.2	4.0	6.1	6.0	6.8	6.0
GM5	2.0	4.0	4.1	5.0	7.5	8.1	9.4	10.7
KO5	4.2	7.6	8.0	10.0	13.0	14.5	16.9	18.0
BF6	3.0	6.6	6.5	7.9	10.8	12.3	13.5	13.8
DH6	2.5	4.6	4.2	5.7	7.4	8.5	9.5	9.5
FL6	3.0	5.1	4.8	6.6	9.8	10.1	11.8	12.6
HN6	3.0	5.6	5.8	7.1	10.2	11.0	12.4	13.6
BF7	3.4	6.5	5.5	7.6	9.9	11.7	12.7	14.0
EK7	2.3	4.9	3.9	6.1	8.1	8.7	9.0	10.0
HN7	2.7	6.6	6.2	7.3	9.4	10.9	11.7	13.7
AE8	4.5	7.6	8.7	10.6	12.5	14.0	14.6	16.5
BF8	3.2	6.3	7.4	8.8	11.1	12.8	13.5	14.9
CG8	2.9	5.9	6.9	7.4	10.7	12.1	13.0	14.0
DH8	2.0	4.9	5.1	6.1	8.0	10.1	10.4	12.0
EK8	1.8	4.5	5.0	5.8	7.5	10.4	10.5	10.6
FL8	3.2	5.7	6.0	7.2	9.1	11.7	11.9	12.1
GM8	3.5	6.4	5.7	7.3	9.6	10.6	10.5	11.3
HN8	3.0	6.3	6.5	7.6	10.6	12.0	12.6	14.0
KO8	3.2	6.5	6.8	8.1	11.1	12.3	13.0	15.0
BF9	3.9	6.9	7.8	8.8	12.7	14.1	14.8	16.0
EK9	2.5	4.9	5.0	6.0	9.1	9.9	10.4	11.3
HN9	3.0	6.1	7.0	7.6	9.8	11.2	12.7	14.1
BF10	1.8	4.7	5.0	5.8	8.2	10.7	11.1	12.9
DH10	1.5	4.0	4.0	4.9	5.7	6.7	8.0	9.2
FL10	2.2	4.8	5.7	6.4	7.5	8.6	9.3	10.4
HN10	2.5	6.4	6.5	8.5	9.7	11.3	12.4	13.6
AE11	4.0	6.9	8.0	10.0	12.1	14.0	15.0	17.5
CG11	2.8	4.2	4.3	5.3	7.3	8.3	9.1	10.7
EK11	2.0	3.4	3.5	5.0	5.5	5.7	5.4	5.5
GM11	2.9	5.1	5.9	7.4	8.8	10.1	10.1	10.9
KO11	3.9	6.4	7.3	9.9	11.9	13.1	14.3	15.7
EK13	1.0	1.2	1.4	2.3	2.4	2.4	2.1	2.4
DH15	.2	.4	.4	1.3	1.2	1.0	1.2	1.2
FL15	-.5	.4	.4	1.2	1.3	1.3	1.3	1.7

TABLE VI
CROSS BENDING TESTS
(Continued)

Gage Lines	Load in Lbs. on Each Arm 90	165	190	215	265	290	305	320
6-10B	-.3	.7	.3	1.2	1.0	1.4	1.1	1.3
2-6D	3.3	7.0	7.5	9.2	11.4	13.0	13.4	14.9
4-8D	1.8	3.6	4.1	5.6	6.8	7.3	8.4	9.3
6-10D	1.5	3.2	3.1	5.0	5.7	7.0	6.8	6.5
8-12D	2.1	4.9	5.1	6.1	7.7	9.0	9.1	11.7
10-14D	3.9	7.1	8.0	10.1	12.2	14.3	14.8	17.1
3-7E	3.0	5.9	6.3	8.0	9.9	11.0	11.5	12.9
5-9E	2.5	4.4	4.8	5.0	6.6	7.4	8.0	9.4
7-11E	2.3	4.2	4.6	5.6	7.0	7.9	8.2	9.3
9-13E	3.1	6.1	6.7	8.0	9.5	11.3	12.0	14.6
3-7F	3.9	6.1	7.2	9.1	11.2	12.1	13.0	14.5
6-10F	1.6	3.0	3.7	5.0	6.6	7.6	8.4	8.9
9-13F	3.1	6.0	7.1	8.0	10.0	11.8	12.8	13.9
1-5G	2.1	5.0	6.3	7.1	9.2	10.8	11.9	13.0
2-6G	3.0	6.0	6.8	8.9	10.7	12.0	12.6	13.3
3-7G	2.7	5.8	6.5	8.2	9.9	11.0	11.8	12.8
4-8G	2.6	5.0	6.1	7.1	10.0	10.2	11.0	12.0
5-9G	1.8	4.8	6.6	7.8	9.8	10.7	11.7	11.8
6-10G	1.3	4.4	5.0	6.1	7.4	9.2	10.2	10.7
7-11G	1.6	3.4	3.7	4.9	6.1	9.3	9.1	10.1
8-12G	3.0	5.9	7.4	7.9	10.0	12.2	13.0	13.9
9-13G	3.7	6.1	7.3	8.9	11.2	12.3	13.0	14.0
10-14G	3.7	7.0	8.0	9.6	11.9	13.3	14.5	15.8
11-15G	5.8	9.6	10.7	12.5	14.8	17.0	17.6	18.4
3-7H	4.1	5.8	6.5	8.1	11.0	12.3	13.0	14.0
6-10H	3.1	6.1	6.5	6.8	9.0	10.7	12.0	13.0
9-13H	3.1	5.8	6.7	7.9	9.9	11.7	11.9	13.4
3-7K	2.0	3.9	5.0	5.9	7.6	9.1	9.9	10.9
5-9K	3.0	5.0	6.1	6.6	8.1	9.2	10.1	11.1
7-11K	3.0	4.8	5.1	6.1	7.1	8.1	9.0	10.6
9-13K	2.9	5.3	6.4	7.4	9.3	11.0	12.0	14.1
2-6L	3.5	6.3	7.3	9.3	12.0	13.5	14.6	16.8
4-8L	2.6	4.6	6.2	7.4	9.5	9.6	11.0	12.4
6-10L	1.9	2.9	3.0	4.2	5.0	4.8	5.7	5.6
8-12L	2.3	4.1	4.2	5.2	6.7	8.1	9.0	10.2
10-14L	4.0	6.6	8.2	9.5	11.6	13.6	14.7	16.1
6-10N	.8	1.7	1.8	2.7	2.5	3.0	3.5	3.0
1-2T	3.9	6.7	8.0	10.2	11.9	12.9	14.7	16.0
3-4T	3.6	7.1	7.9	8.8	11.5	13.6	15.1	16.6

TABLE VII
Theoretical Stress in
Plates in Cross
Bending Tests

$$\frac{f\ I}{C} = M$$

First Series		Second Series	
Load Per Arm Lbs	Stress Per Sq. In. Lbs.	Load Per Arm Lbs.	Stress Per Sq. In. Lbs.
32.5	4000	90	11100
57.5	7100	165	20300
82.5	10100	190	23400
107.5	13200	215	26400
132.5	16200	265	32500
157.5	19300	290	35600
182.5	22400	305	37400
207.5	25500	320	39300
232.5	28500		
265	32500		
290	35600		
315	38700		
340	41700		
365	44800		
375	46000		

TABLE VIII
Thickness of Walls of Tube
No. 1
Decimals of an Inch

	1	1a	2	2a	3	3a	4	4a
A1	.086	.095	.093	.093	.096	.093	.089	.085
A2	.084	.092	.095	.094	.096	.094	.087	.082
A3	.082	.092	.096	.094	.096	.093	.087	.081
B1	.082	.093	.095	.092	.093	.094	.086	.079
B2	.082	.092	.096	.092	.094	.096	.088	.077
C1	.080	.092	.094	.091	.094	.095	.088	.076
C2	.079	.091	.094	.091	.094	.095	.087	.073
D1	.078	.090	.094	.091	.092	.094	.085	.072
D2	.078	.090	.095	.092	.092	.095	.087	.073
E1	.078	.089	.094	.091	.092	.095	.087	.074
E2	.078	.087	.093	.092	.094	.097	.088	.077
E3	.079	.087	.094	.093	.096	.097	.090	.079

TABLE IX
Thickness of Walls of Tube
No. 2

	1	1a	2	2a	3	3a	4	4a
A1	.088	.088	.088	.090	.088	.086	.090	.090
A2	.085	.083	.087	.089	.086	.083	.090	.089
A3	.084	.083	.088	.087	.085	.085	.091	.090
B1	.083	.084	.089	.087	.083	.085	.090	.089
B2	.084	.088	.091	.089	.085	.085	.090	.089
C1	.085	.090	.091	.087	.083	.086	.090	.088
C2	.085	.088	.090	.087	.082	.087	.090	.088
D1	.085	.089	.092	.087	.085	.089	.091	.087
D2	.085	.090	.092	.088	.087	.090	.091	.088
E1	.086	.090	.094	.088	.087	.088	.091	.088
E2	.088	.092	.096	.088	.087	.087	.091	.090
E3	.090	.093	.096	.088	.087	.087	.091	.093

TABLE X
Thickness of Walls of Tube
No. 3

	1	1a	2	2a	3	3a	4	4a
A1	.091	.085	.086	.089	.089	.090	.090	.092
A2	.092	.084	.086	.088	.088	.087	.089	.094
A3	.091	.084	.083	.084	.088	.087	.089	.095
B1	.091	.084	.080	.085	.088	.087	.091	.096
B2	.091	.084	.084	.086	.088	.085	.089	.094
C1	.090	.084	.084	.087	.086	.084	.089	.093
C2	.090	.087	.086	.087	.082	.082	.090	.093
D1	.088	.086	.088	.084	.078	.083	.091	.092
D2	.087	.087	.091	.084	.079	.085	.092	.091
E1	.085	.089	.094	.082	.078	.085	.092	.088
E2	.085	.091	.094	.083	.077	.084	.091	.088
E3	.085	.092	.092	.082	.077	.082	.090	.087

TABLE XI
Thickness of Walls of Tube
No. 4

	1	1a	2	2a	3	3a	4	4a
A1	.085	.085	.091	.091	.089	.090	.094	.091
A2	.084	.085	.091	.091	.086	.086	.092	.090
A3	.083	.084	.091	.090	.083	.084	.090	.089
B1	.082	.084	.090	.088	.080	.083	.089	.088
B2	.081	.083	.088	.086	.081	.084	.088	.087
C1	.081	.084	.087	.084	.082	.084	.086	.083
C2	.082	.086	.087	.083	.082	.083	.083	.080
D1	.084	.087	.087	.083	.081	.083	.083	.079
D2	.084	.087	.086	.081	.080	.082	.082	.081
E1	.085	.087	.086	.083	.082	.083	.084	.083
E2	.085	.085	.086	.083	.084	.084	.083	.084
E3	.085	.084	.085	.083	.083	.082	.081	.083

TABLE XII
Thickness of Walls of Tube
No. 5

	1	1a	2	2a	3	3a	4	4a
A1	.087	.092	.095	.092	.088	.093	.097	.092
A2	.084	.090	.097	.092	.087	.094	.098	.092
A3	.081	.091	.097	.093	.087	.094	.098	.090
B1	.080	.093	.098	.093	.088	.092	.097	.089
B2	.083	.093	.098	.092	.087	.092	.097	.090
C1	.083	.093	.099	.092	.086	.093	.099	.092
C2	.083	.094	.099	.091	.087	.094	.099	.093
D1	.082	.094	.099	.090	.086	.095	.100	.090
D2	.082	.094	.099	.090	.086	.096	.099	.088
E1	.083	.095	.098	.090	.084	.097	.097	.088
E2	.086	.095	.099	.089	.086	.099	.100	.089
E3	.087	.094	.096	.088	.089	.099	.099	.090

TABLE XIII
Thickness of Walls of Tube
No. 6

	1	1a	2	2a	3	3a	4	4a
A1	.086	.082	.082	.084	.084	.086	.083	.087
A2	.085	.079	.081	.082	.082	.084	.082	.086
A3	.087	.081	.078	.080	.083	.085	.083	.088
B1	.087	.080	.079	.079	.082	.086	.083	.085
B2	.087	.081	.078	.078	.082	.085	.085	.085
C1	.085	.081	.079	.078	.082	.087	.084	.084
C2	.083	.080	.079	.080	.082	.087	.086	.083
D1	.079	.078	.079	.079	.081	.087	.083	.082
D2	.079	.079	.079	.078	.080	.084	.083	.082
E1	.079	.078	.079	.078	.079	.083	.083	.079
E2	.077	.078	.079	.080	.079	.081	.082	.079
E3	.078	.080	.080	.079	.079	.082	.081	.079

TABLE XIV
Thickness of Walls of Tube
No. 7

	1	1a	2	2a	3	3a	4	4a
A1	.093	.094	.094	.093	.095	.092	.091	.094
A2	.093	.094	.092	.095	.094	.092	.091	.095
A3	.092	.092	.093	.094	.093	.091	.092	.095
B1	.091	.090	.093	.094	.091	.090	.093	.095
B2	.090	.088	.094	.093	.090	.088	.093	.094
C1	.090	.088	.093	.093	.090	.089	.094	.095
C2	.088	.089	.093	.092	.091	.089	.092	.094
D1	.089	.089	.092	.091	.091	.090	.093	.093
D2	.090	.090	.091	.091	.090	.091	.092	.092
E1	.089	.090	.092	.090	.089	.093	.092	.092
E2	.091	.090	.090	.092	.092	.092	.093	.091
E3	.090	.091	.091	.091	.092	.092	.092	.090

TABLE XV
Thickness of Walls of Tube
No. 8

	1	1a	2	2a	3	3a	4	4a
A1	.093	.096	.100	.097	.092	.093	.094	.095
A2	.092	.096	.101	.099	.091	.092	.092	.094
A3	.091	.096	.100	.099	.090	.092	.093	.093
B1	.093	.098	.102	.100	.093	.092	.094	.093
B2	.093	.097	.100	.097	.094	.092	.095	.092
C1	.093	.096	.101	.097	.093	.093	.094	.092
C2	.093	.096	.102	.096	.093	.093	.095	.092
D1	.094	.098	.099	.096	.094	.095	.098	.093
D2	.096	.098	.099	.097	.096	.097	.098	.094
E1	.097	.099	.099	.098	.097	.098	.097	.096
E2	.099	.098	.099	.100	.099	.098	.098	.097
E3	.099	.098	.102	.101	.101	.099	.098	.098

TABLE XVI
Thickness of Walls of Tube
No. 9

	1	1a	2	2a	3	3a	4	4a
A1	.096	.099	.102	.098	.097	.101	.104	.099
A2	.096	.098	.101	.097	.096	.101	.102	.096
A3	.096	.097	.099	.099	.096	.099	.103	.095
B1	.096	.097	.097	.100	.097	.102	.102	.096
B2	.095	.096	.095	.098	.099	.102	.103	.096
C1	.094	.093	.095	.097	.099	.101	.101	.094
C2	.094	.093	.094	.095	.099	.102	.101	.095
D1	.093	.093	.092	.096	.100	.104	.101	.095
D2	.092	.093	.091	.096	.100	.104	.101	.094
E1	.094	.093	.091	.094	.100	.103	.100	.094
E2	.095	.093	.092	.096	.100	.102	.100	.095
E3	.096	.095	.093	.098	.100	.101	.101	.098

[illegible]

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE XVII
CALIBRATIONS OF
HYDRAULIC GAGES

Scale Reading	1000 Lb. Gage Gage Reading *	#
50	55	55
100	105	105
125		130
150	160	155
175		180
200	210	210
225	235	235
250	260	260
275	287	288
300	315	312
325		340
350	365	365
375		393
400	420	420
425		445
450	470	473
475		497
500	525	525
550	575	
600	625	
650	680	
700	730	
750	780	
800	835	
850	885	
900	935	
950	980	

Scale Reading	2000 Lb. Gage Gage Reading *
100	100
200	200
300	300
400	400
500	500
600	600
700	700
800	800
900	900
1000	1000
1100	1110
1200	1210
1300	1310
1400	1415
1500	1520
1600	1620
1700	1720
1800	1820

* Calibrated on 1500 lb. Crosby gage tester in M. E. Laboratory.

Calibrated on Crosby Dead weight gage tester.

Calibrations up and down coincided.

TABLE XVIII
Tension Test of Specimens Cut From
Tubes 2 and 3
Unit Elongation Taken
From Average Curves

Unit Stress	Unit Elongation				General Average
5000	.00014	.00014	.00014	.00015	.00014
10000	.00027	.00028	.00028	.00028	.00028
15000	.00040	.00042	.00042	.00042	.00042
20000	.00054	.00056	.00057	.00057	.00056
22500	.00063	.00064	.00064	.00064	.00064
25000	.00072	.00073	.00072	.00072	.00072
27500	.00080	.00081	.00079	.00081	.00080
30000	.00089	.00090	.00085	.00089	.00089
32500	.00098	.00099	.00093	.00099	.00098
35000	.00107	.00108	.00100	.00107	.00106
37500	.00115	.00117	.00107	.00115	.00114
40000	.00125	.00126	.00118	.00123	.00123
42500	.00138	.00134	.00129	.00134	.00134
45000	.00160	.00143	.00142	.00146	.00148
47500	.00190	.00156	.00160	.00157	.00165
50000	.00226	.00172	.00182	.00172	.00189

Ewing telescopic Extensometer used.
Gage length 1.266 inches.

TABLE XVIIIA
Tension Tests of Specimens Cut From
Tubes 6, 7, and 8
Unit Elongation Taken
From Average Curves

Unit Stress	Unit Elongation				General Average
5000	.00014	.00015	.00015	.00014	.00015
10000	.00028	.00031	.00033	.00028	.00031
12500	.00036	.00038	.00041	.00035	.00037
15000	.00042	.00046	.00048	.00041	.00045
17500	.00050	.00054	.00057	.00048	.00052
18500	.00056	.00057	.00061	.00051	.00056
20000	.00063	.00061	.00065	.00057	.00062
21000	.00068	.00064	.00069	.00059	.00065
22500	.00079	.00069	.00074	.00068	.00073
23500	.00089	.00074	.00077	.00073	.00079
25000	.00112	.00084	.00084	.00080	.00090
26000	.00125	.00093	.00090	.00083	.00098

Average of two tests for yield point by drop of beam, 23,200 lbs. per square inch.

Ewing telescopic Extensometer used.
Gage length 1.266 inches.

TABLE XIX
Tension Tests of Specimens Cut
From Same Plate as the Cross
Bending Specimens

Average of Two

Unit Stress	Unit Elongation
8180	.00017
16500	.00061
24400	.00090
28400	.00130
31600	.00370
32500	.00650

Specimen .243" x 1.009"
Area = .2396 Sq. in.
.233 x 1.002
Area .2279 Sq. in.

Average of 3 yield points by drop of Beam

34300
35300
36500

Average 35400 Lbs per Sq. In.

Ultimate Limit

57100
56900
57400
59000
57300

Average 57500

TABLE XX
Computed and General Average of Observed Deformations
Axial Gage Lines
Unit Deformation x 10,000

E from Tube No. 5 - 27,300,000
Poisson's ratio from Tube No. 5 - .334

Unit Stress	Axial	Circum-ferential	Averaged Observed Deformations				Computed Deformations				
	Tube No. 5		Tube Number 1	2	4	3	1	2	4	3	Stress/E
5000	1.9	-.6	1.6	1.5	1.4	1.3	1.7	1.6	1.4	1.3	1.8
10000	3.7	-1.3	3.1	2.9	2.9	2.6	3.4	3.1	2.9	2.6	3.7
20000	7.4	-2.5	6.2	5.8	5.5	5.2	6.8	6.2	5.7	5.1	7.3
25000	9.2	-3.1	7.7	7.2	6.9	6.5	8.5	7.8	7.1	6.4	9.1
30000	11.0	-3.7	9.3	8.6	8.3	7.8	10.2	9.3	8.6	7.7	11.0
35000	13.0	-4.3	11.4	10.1	9.8	9.2	11.9	10.9	10.0	9.0	12.8
40000	15.1	-5.0	13.4	11.5	11.5	10.7	13.5	13.4	11.3	10.2	14.6
42500	16.2	-5.3	14.4	12.2	12.3	11.5	14.3	13.2	12.0	10.8	15.5
45000	17.8	-5.8	15.5	12.8	13.2	12.3	15.2	14.0	12.7	11.5	16.4
47500	19.7	-6.3	16.6	13.8	14.1	13.1	16.1	14.7	13.4	12.1	17.4
50000	22.4	-6.9	18.0	14.8	15.1	14.0	16.9	15.5	14.2	12.8	18.3
52500		-7.5	19.5	15.9	16.3	15.1		16.3	14.9	13.4	19.2
55000			21.1	17.4	18.0	17.3					
57500			23.0	19.0	20.1	21.2					
60000			25.3	21.0	24.9						
62500			28.6	23.4							

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TABLE XXI
Computed and General Average of Observed Deformations
Axial Gage Lines
Unit Deformation x 10,000

Unit Stress	Average observed deformations Tube Number 7	6	8	Computed deformations Tube Number 7	6 and 8
2500	.9	.7	.7	.8	.7
5000	1.8	1.3	1.4	1.6	1.3
10000	3.6	2.6	2.7	3.1	2.6
12500	4.5	3.4	3.4	3.9	3.3
15000	5.4	4.0	4.1	4.7	3.9
17250	6.3	4.6	4.8	5.4	4.5
20000	7.3	5.3	5.5	6.2	5.1
21250	7.7	5.6	5.9	6.6	5.4
22500	8.1	5.9	6.2	7.0	5.8
23750	8.6	6.3	6.6	7.4	6.1
25000	9.1	6.6	7.0	7.8	6.5
26250	9.7	7.1	7.3	8.2	6.8
27250	10.6	7.6	7.8	8.6	7.1
28750	11.8	8.3	8.4	9.0	7.4
30000	13.0	9.0	9.3	9.3	7.7
31250	14.3	9.9	10.4		
32500	15.8	11.4	11.7		
33750		16.3	13.2		

TABLE XXII
Computed and General Average of Observed Deformations
Circumferential Gage Lines
Unit Deformation x 10,000

Unit Stress	Average Observed Deformations					Computed Deformations			
	Tube Number								
	5	1	2	4	3	1	2	4	3
5000	-.6	-.9	.5	.7	1.0	-.7	.5	.9	1.2
6250		-1.1				-.9			
7500	-.9	-1.3				-1.1			
8750	-1.0	-1.6				-1.3			
10000	-1.3	-1.8	1.0	1.4	2.1	-1.4	1.1	1.9	2.4
11250	-1.4	-2.0				-1.6			
12500	-1.5	-2.3				-1.8			
13750		-2.6				-2.0			
15000	-1.9	-3.2	1.4	2.0	3.1		1.6	2.8	3.6
20000	-2.5		1.9	2.6	4.1		2.1	3.7	4.8
22500				2.9			2.4	4.2	
25000	-3.1		2.3	3.3	5.1	-3.6	2.7	4.7	5.9
27500				3.7			3.0	5.1	
30000	-3.7		2.8	4.0	6.1		3.2	5.6	7.1
32500				4.4			3.5	6.1	
35000	-4.3			4.8	7.1		3.8	6.6	8.3
37500				5.2			4.1	7.1	
40000	-5.0			5.6	8.2				9.5
41250	-5.1			6.0					
42500	-5.3			6.8					
45000	-5.8				9.2				10.7
47500									
50000	-6.9				10.2				11.9
52500	-7.5				11.7				
53750					14.0				

TABLE XXIII
Computed and Average of
Observed Deformations
Circumferential Gage Lines
Unit Deformation x 10,000

Unit Stress	Average observed deformations			Computed deformations	
	Tube Numbers				
	7	6	8	7	6 and 8
2500	.3	.4	.4	.3	.6
5000	.7	.9	.8	.5	1.2
6250	.9	1.1	1.0	.7	1.5
7500	1.0	1.3	1.2	1,00	1.8
8750	1.2	1.5	1.5	.10	2.1
10000	1.4	1.7	1.7	1.1	2.4
11250	1.5	1.9	1.9	1.3	2.7
12500	1.7	2.1	2.1	1.4	3.0
13750	1.8	2.3	2.3	1.5	3.3
15000	1.8	2.5	2.6	1.6	3.6
17500		3.0	3.1		4.2
20000		3.5	3.6		4.8
22500		4.1	4.0		5.4
25000		4.7	4.5		5.9
27500		5.6	5.3		6.5
30000		7.6	6.6		
32500			8.7		
34500			10.8		

TABLE XXIV.

TUBE NO1

Ratio of Hoop to Axial Tension 0.24

Inside Diameter of Tube 5.564 inches

Axial Gage Lines

Section AB

Average Thickness of Tube, .089 in. Area of Section, 1.590 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av. Axial Unit Stress lbs. sq. in.	Reading of Gage Lines				Differences				Average Diff.	Average Unit Elongation
Gage Rdg	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
50	46	1100	9100	10200	6400	7.0	15.0	93.1	0.1	0	0	0	0	0	0
150	145	3500	26100	29600	18600	99.1	8.9	86.0	90.9	7.9	6.1	7.1	9.2	[illegible]	.00038
225	215	5200	38800	44000	27700	92.9	2.0	81.0	93.8	14.1	13.0	12.1	16.3	13.9	.00070
275	263	6400	47400	53800	33900	88.0	98.7	77.1	79.8	19.0	16.3	16.0	20.3	17.9	.00090
300	285	6900	51600	58500	36800	85.1	96.1	74.7	77.1	21.9	18.9	18.4	23.0	20.6	.00103
325	310	7500	55900	63400	39900	82.8	94.8	72.9	74.8	24.2	20.2	20.2	25.3	22.5	.00113
350	335	8100	60100	68100	42900	80.0	92.9	70.0	72.0	27.0	22.1	23.1	28.1	25.1	.00126
375	357	8700	64400	73100	46000	77.9	90.9	68.0	69.8	29.1	24.1	25.1	30.3	27.2	.00136
400	380	9200	68700	77900	49000	75.0	89.0	65.9	67.0	32.0	26.0	27.2	33.1	29.6	.00148
425	405	9900	72900	82800	52100	72.1	87.0	63.8	64.9	34.9	28.0	29.3	35.2	31.9	.00160
450	430	10500	77200	87700	55200	70.0	85.0	61.0	62.0	37.0	30.0	32.1	38.1	34.3	.00172
475	453	11000	81400	92400	58100	66.5	82.8	58.6	59.1	40.5	32.2	34.5	41.0	37.1	.00186
500	475	11500	85600	97100	61100	60.9	78.1	54.4	53.8	46.1	36.9	38.7	46.3	42.0	.00210
525	500	12200	89900	102100	64300	35.0	67.0	50.0	35.0	72.0	48.0	43.1	65.1	57.1	.00286

Section BC

Average Thickness of Tube, .088 in. Area of Section, 1.572 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av Axial Unit Stress lbs sq. in.	Reading of Gage Lines				Differences				Average Diff.	Average Unit Elongation
Gage Rdg.	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
50	46	1100	9100	10200	6500	89.1	60.7	86.1	92.5	0	0	0	0	0	0
150	145	3500	26100	29600	18800	81.0	53.6	79.8	84.0	8.1	7.1	6.3	8.5	7.5	.00038
225	215	5200	38800	44000	28000	74.3	47.0	73.9	77.2	14.8	13.7	12.2	15.3	14.0	.00070
275	263	6400	47400	53800	34200	69.0	43.0	69.1	73.0	20.1	17.7	17.0	19.5	18.6	.00093
300	295	6900	51600	58500	37300	66.9	40.4	67.4	71.0	22.2	20.3	18.7	21.5	20.7	.00104
325	310	7500	55900	63400	40400	64.4	37.0	66.0	68.2	24.7	23.7	20.1	24.3	23.2	.00116
350	335	8100	60100	68200	43500	61.0	33.7	63.9	66.7	28.0	27.0	22.2	25.8	25.8	.00129
375	357	8700	64400	73100	46600	57.8	29.2	61.8	64.2	31.3	31.5	24.3	28.3	29.4	.00147
400	380	9200	68700	77900	49700	53.9	24.9	59.5	61.8	35.2	35.8	26.6	30.7	32.1	.00161
[illegible]	405	9900	72900	82800	52700	50.1	19.2	57.4	59.6	39.0	41.5	28.7	32.9	35.3	.00177
[illegible]	430	10500	77200	87700	55900	45.8	13.0	54.8	57.3	43.3	47.7	31.3	35.2	39.4	.00187
475	453	11000	81400	92400	58800	40.0	6.0	52.3	53.5	49.1	54.7	33.8	39.0	44.2	.00221
500	475	11500	85600	97100	61900	31.0	92.1	48.0	49.9	58.1	68.6	38.1	42.6	51.9	.00260
525	500	12200	89900	102100	65000	21.2	78.1	19.0	34.3	67.9	82.6	67.1	58.2	69.0	.00345

Section CD

Average Thickness of Tube .086 in. Area of Section 1.527 sq. in.

Water Pressure		Axial Load due to W Pressure	Machine Load lbs.	Total Axial Load lbs.	Av Axial Unit Stress lbs sq in	Reading on Gage Line				Differences				Av Diff.	Av Unit Elongation
Gage Rdg	Corrected.					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
50	46	1100	9100	10200	6700	88.1	9.0	95.8	98.8	0	0	0	0	0	0
[illegible]	145	3500	26100	29600	19300	80.0	[illegible]	88.1	91.0	8.1	7.1	7.7	7.8	7.7	.00039
225	215	5200	38800	44000	28800	73.2	[illegible]	[illegible]	84.0	14.9	12.4	12.8	14.8	13.2	.00066
275	263	6400	47400	53800	35200	68.0	[illegible]	73.9	79.4	20.1	16.8	16.9	19.4	18.3	.00092
300	285	6900	51600	58500	[illegible]	65.0	89.1	[illegible]	77.5	23.1	19.9	18.8	21.3	20.8	.00104
325	310	7500	55900	63400	[illegible]	61.5	86.0	[illegible]	74.8	26.3	[illegible]	21.3	24.0	23.7	.00119
350	335	8100	60100	68200	44700	58.0	82.8	72.3	72.7	30.1	[illegible]	23.5	26.1	26.5	.00133
375	357	8700	64400	73100	47900	53.0	78.8	70.6	70.3	35.1	30.2	25.2	28.5	30.0	.00150
400	380	9200	68700	77900	51200	48.0	74.0	68.0	67.8	40.1	35.0	27.8	31.0	33.5	.00168
425	405	9900	72900	82800	54300	43.0	68.0	65.9	65.8	45.1	41.0	29.9	33.0	37.3	.00187
450	430	10500	77200	87700	57400	35.5	64.0	62.8	63.3	52.6	45.0	33.0	35.5	41.5	.00208
475	453	11000	81400	92400	60600	27.1	57.8	59.8	59.9	61.0	51.2	36.0	38.9	46.8	.00234
500	475	11500	85600	97100	63800	10.9	50.4	55.5	52.0	77.2	58.6	40.3	46.8	55.7	.00279
525	500	12200	89900	102100	67000	—	31.0	46.5	26.0	—	78.0	49.3	72.8	—	—

TABLE XXV.

TUBE NO. 2

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.558 inches

Axial Gage Lines

TABLE XXV.

TUBE NO. 2

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.558 inches

Axial Gage Lines

Section A B

Average Thickness of Tube .087 in Area of Section 1.543 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Lines				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg.	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	95	2300	7900	10200	6600	89.9	16.5	34.0	93.7	0.0	0.0	0.0	0.0	0.0	0
300	285	6900	22400	29300	19000	80.9	7.4	26.0	85.0	9.0	9.1	8.0	8.7	8.7	.00044
500	475	11500	37000	48500	31500	73.5	0.0	19.1	77.5	16.4	16.5	14.9	16.2	16.0	.00080
600	575	14000	44300	58300	37800	69.0	96.3	15.0	73.6	20.9	20.2	19.0	20.1	20.1	.00101
700	670	16300	51600	67900	44000	65.5	93.0	12.9	69.0	24.4	23.5	21.1	24.7	23.4	.00117
800	765	18600	58800	77200	50100	61.1	88.0	7.5	63.0	28.8	28.5	26.5	30.7	28.6	.00143
850	815	19800	62400	82200	53300	58.8	86.1	5.0	59.8	31.1	30.4	29.0	33.9	31.1	.00156
900	865	21000	66100	87100	56500	55.5	83.9	1.5	55.3	34.9	32.6	32.5	38.4	34.6	.00173
950	915	22200	69800	92000	59600	50.5	79.9	95.4	47.5	39.4	36.6	38.6	46.2	40.2	.00201
1000	965	23400	73400	96800	62800	46.0	70.0	92.0	44.0	43.9	46.5	42.0	49.1	45.5	.00228

Section BC

Average Thickness of Tube .087 in Area of Section 1.543 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Lines				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg.	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
100	95	2300	7900	10200	6600	83.9	24.1	46.0	0.9	0	0	0	0	0	0
300	285	6900	22400	29300	19000	77.9	16.9	40.8	97.0	6.0	7.2	5.2	3.9	5.6	.00028
500	475	11500	37000	48500	31500	68.5	11.8	34.0	90.0	15.4	12.3	12.0	10.9	12.7	.00064
600	575	14000	44300	58300	37800	66.1	9.9	32.0	85.5	17.8	14.2	14.0	15.4	15.4	.00077
700	670	16300	51600	67900	44000	61.8	5.7	29.1	80.1	22.1	18.4	16.9	20.8	19.6	.00098
800	765	18600	58800	77200	50100	57.5	1.6	24.0	73.9	26.4	22.5	22.0	27.0	24.5	.00123
850	815	19800	62400	82200	53300	55.6	0.0	21.5	71.1	28.3	24.1	24.5	29.8	26.7	.00134
900	865	21000	66100	87100	56500	52.5	95.9	16.9	64.5	31.4	28.2	29.1	36.4	31.3	.00157
950	915	22200	69800	92000	59600	47.0	90.0	11.1	56.5	36.9	34.1	34.9	44.4	37.6	.00188
1000	965	23400	73400	96800	62800	44.5	88.0	7.8	54.8	39.4	36.1	38.2	46.1	40.0	.00200

Section C D

Average Thickness of Tube .0885 in. Area of Section 1.570 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Lines				Differences.				Av Diff.	Av. Unit Elongation
Gage Rdg	Corrected					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
100	95	2300	7900	10200	6570	15.0	69.1	53.0	99.4	0	0	0	0	0	0
300	285	6900	22400	29300	18700	9.0	61.1	48.0	90.5	6.0	8.0	5.0	8.9	7.0	.00035
500	475	11500	37000	48500	30900	1.0	54.1	43.0	84.5	14.0	15.0	10.0	14.9	13.5	.00068
600	575	14000	44300	58300	37200	97.5	50.5	39.0	79.9	17.5	18.6	14.0	19.5	17.4	.00087
700	670	16300	51600	67900	43300	93.9	47.5	34.9	76.1	21.1	21.6	18.1	23.3	21.0	.00105
800	765	18600	58800	77200	49200	89.5	42.2	31.9	68.1	25.5	26.9	21.1	31.3	26.2	.00131
850	815	19800	62400	82200	52400	85.6	40.0	28.0	64.1	29.4	28.5	25.0	35.3	29.6	.00148
900	865	21000	66100	87100	55500	82.2	37.5	24.5	61.5	32.8	31.6	28.5	37.9	32.7	.00164
950	915	22200	69800	92000	58600	79.5	32.0	21.1	57.5	35.5	37.1	31.9	41.9	36.6	.00183
1000	965	23400	73400	96800	61700	75.0	26.5	18.5	55.0	40.0	42.6	34.5	44.4	40.4	.00202

TABLE XXVI.

TUBE NO. 3

Rati of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.561 inches

Axial Gage Lines

TABLE XXVI.

TUBE NO. 3

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.561 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .0877 in. Area of Section 1.553 sq. in.

Water Pressure Gage Rdg	Water Pressure Corrected	Axial Load due to W. Pressure	Machine Load lbs	Total Axial Load lbs	Av. Axial Unit Stress lbs sq in	Reading on Gage Lines AB1	AB2	AB3	AB4	Differences AB1	AB2	AB3	AB4	Av. Diff	Av. Unit Elongation
100	100	2425	3600	6025	3900	40.7	50.6	99.5	40.0	0	0	0	0	0	0
400	400	9700	11900	21600	13900	33.9	43.4	95.0	34.8	6.8	7.2	4.5	5.2	5.9	.00030
700	700	17000	20200	37200	24000	29.2	40.0	88.9	29.8	11.5	10.6	10.6	10.2	10.7	.00054
900	900	21800	26000	47800	30800	27.0	37.5	85.0	26.5	13.7	13.1	14.5	13.5	13.7	.00069
1100	1090	26400	31400	57800	37200	19.8	32.3	81.2	22.9	20.9	18.3	18.3	17.1	18.7	.00094
1300	1290	31300	37000	68300	44000	18.0	26.7	77.0	17.5	22.7	23.9	22.5	22.5	22.9	.00115
1400	1390	33700	39800	73500	47400	15.4	24.0	74.0	14.3	25.3	26.6	25.5	25.7	25.8	.00129
1600	1590	38600	45400	84000	54100	12.0	16.0	65.6	8.0	28.7	34.6	33.9	32.0	32.3	.0016[illegible]
1700	1690	41000	48200	89200	57500	0.0	3.2	57.7	96.0	40.7	47.4	41.8	44.0	43.5	.00218
1750	1740	42200	49200	91400	58900	49.0	53.0	27.0	46.0	91.7	97.6	72.5	94.0	89.0	.004[illegible]

Section BC

Average Thickness of Tube .0871 in Area of Section 1.544 sq. in.

Water Pressure Gage Rdg.	Water Pressure Corrected	Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av Axial Unit Stress lbs. sq. in	Reading on Gage Line BC1	BC2	BC3	BC4	Differences BC1	BC2	BC3	BC4	Av Diff.	Av Unit Elongation
100	100	2425	3600	6025	3900	49.0	18.3	58.7	28.5	0	0	0	0	0	.0
[illegible]	400	9700	11900	21600	14000	45.5	11.5	51.9	22.8	3.5	6.8	6.8	5.7	5.7	.00029
700	700	17000	20200	37200	24100	41.0	6.5	47.5	16.9	8.0	11.8	11.2	11.6	10.7	.00054
900	900	21800	26000	47800	31000	36.8	2.5	43.8	13.0	12.2	15.8	14.9	15.5	14.6	.00073
1100	1090	26400	31400	57800	37500	33.4	99.8	40.4	10.4	15.6	18.5	18.3	18.1	17.6	.00088
1300	1290	31300	37000	68300	44200	29.5	95.5	36.5	5.9	19.5	22.8	22.2	22.6	21.8	.00109
1400	1390	33700	39800	73500	47700	27.4	93.7	33.7	4.5	21.6	24.6	25.0	24.0	23.8	.00119
1600	1590	38600	45400	84000	54500	21.0	85.0	27.0	98.0	28.0	33.3	31.7	30.5	30.9	.00155
1700	1690	41000	48200	89200	57800	13.0	74.0	19.0	93.8	36.0	44.3	39.7	34.7	38.7	.00194
1750	1740	42200	49200	91400	59300	76.0	14.0	78.0	78.0	73.0	104.3	80.7	50.5	77.1	.00386

Section CD

Average Thickness of Tube .0866 in. Area of Section 1.535 sq in.

Water Pressure Gage Rdg.	Water Pressure Corrected	Axial Load due to W. Pressure	Machine Load lbs	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line CD1	CD2	CD3	CD4	Differences CD1	CD2	CD3	CD4	Av. Diff	Av Unit Elongation
100	100	2425	3600	6025	3900	21.9	40.5	46.9	29.0	0	0	0	0	0	0
400	400	9700	11900	21600	14100	18.2	35.5	42.4	24.9	3.7	5.0	4.5	4.1	4.3	[illegible]22
700	700	17000	20200	37200	24200	13.0	30.0	37.0	19.4	8.9	10.5	9.9	9.6	9.7	.00049
900	900	21800	26000	47800	31200	10.0	26.8	33.2	16.1	11.9	13.7	13.7	12.9	13.1	.00066
1100	1090	26400	31400	57800	37700	6.3	23.0	30.0	12.4	15.6	17.5	16.9	16.6	16.7	.00084
1300	1290	31300	37000	68300	44500	1.6	17.5	25.8	8.0	20.3	23.0	21.1	21.0	21.4	.00107
1400	1390	33700	39800	73500	47900	99.3	14.5	24.0	6.8	22.6	26.0	22.9	22.8	23.6	.00118
1600	1590	38600	45400	84000	54800	93.0	4.9	18.8	1.7	28.9	35.6	28.1	27.3	30.0	.00150
1700	1690	41000	48200	89200	58200	79.3	92.5	8.0	87.0	42.6	48.0	38.9	42.0	42.9	.00215
1750	1740	42200	49200	91400	59600	27.0	60.0	63.0	27.0	94.9	80.5	83.9	102.0	90.3	.00452

TABLE XXVII.

TUBE NO. 4

TABLE XXVII.

TUBE NO. 4

Ratio of Hoop to Axial Tension 0.69

Inside Diameter of Tube 5.563 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .085 in. Area of Section 1.508 sq. in

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff.	Av Unit Elongation
Gage Rdg.	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	100	2430	5100	7500	4950	80.1	10.9	28.0	62.5	0	0	0	0	0.	0
300	300	7290	14100	21400	14200	77.0	5.9	21.9	58.1	3.1	5.0	6.1	4.4	4.7	.00024
500	500	12150	23050	35200	23300	72.5	0.0	17.1	52.7	7.6	10.9	10.9	9.8	9.8	.00049
700	700	17000	32100	49100	32500	66.1	94.0	11.9	47.8	14.0	[illegible]	[illegible]	14.7	15.4	.00077
900	900	21900	41100	63000	41700	61.0	86.9	5.0	41.9	19.1	24.0	23.0	20.6	21.7	.00109
950	950	23100	43400	66500	44000	60.3	84.1	3.5	40.9	19.8	26.8	24.5	21.6	23.2	.00116
1000	1000	24300	45600	69900	46200	59.0	82.0	1.8	39.7	21.1	28.9	26.2	22.8	24.8	.00124
1050	1050	25500	47900	73400	48500	57.3	79.0	99.0	[illegible]	22.1	31.9	29.0	24.6	26.9	.00135
1100	1090	26500	50100	76600	50900	55.2	76.4	97.7	[illegible]	24.9	34.5	30.3	26.4	29.0	.00145
1150	1140	27700	52400	80100	53100	53.1	73.3	94.3	[illegible]	27.0	37.6	32.7	28.5	31.5	.00158
1200	1190	29200	54500	83500	55300	50.9	69.0	93.0	32.1	29.2	41.9	35.0	30.4	34.1	.00171
1250	1240	30100	56800	86900	57600	47.0	65.3	88.0	28.2	33.1	45.6	40.0	34.3	38.3	.00192
1300	1290	31300	59100	90400	60000	30.9	57.9	74.7	16.1	49.2	53.0	53.3	46.4	50.5	.00253

Section BC

Average Thickness of Tube .084 in Area of Section 1.481 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av Diff	Av. Unit Elongation
Gage Rdg.	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
100	100	2430	5100	7500	5060	92.1	79.5	35.0	63.5	0	0	0	0	0	0
300	300	7290	14100	21400	14400	87.0	74.8	29.9	58.4	5.1	4.7	5.1	5.1	5.0	.00025
500	500	12150	23050	35200	23800	81.9	69.0	24.7	53.6	10.2	[illegible]	[illegible]	[illegible]	10.2	.00051
700	700	17000	32100	49100	33100	76.0	63.9	19.1	48.0	16.1	15.6	15.9	15.5	15.8	.00079
900	900	21900	41100	63000	42500	68.9	57.2	13.0	42.9	23.2	[illegible]	[illegible]	20.6	22.0	.00110
950	950	23100	43400	66500	44800	67.2	55.0	11.0	41.2	24.9	24.5	24.0	22.3	23.9	.00120
1000	1000	24300	45600	69900	47200	65.0	53.5	10.3	40.0	27.1	26.0	24.7	23.5	25.4	.00127
1050	1050	25500	47900	73400	49400	62.5	51.5	8.4	38.5	29.6	28.0	26.6	25.0	27.3	.00137
1100	1090	26500	50100	76600	51900	59.9	49.2	6.9	36.6	32.2	30.3	28.1	26.9	29.4	.00147
1150	1140	27700	52400	80100	54100	56.3	47.4	4.8	34.3	35.8	32.1	30.2	29.2	31.8	.00159
1200	1190	29200	54500	83500	56400	52.3	44.8	1.5	30.3	39.8	34.7	33.5	33.2	35.3	.00177
1250	1240	30100	56800	86900	58600	47.8	41.5	97.3	23.1	44.3	38.0	37.7	40.4	40.1	.00201
1300	1290	31300	59100	90400	61100	30.0	30.0	73.8	91.0	62.1	49.2	61.2	72.5	61.3	.00307

Section CD

Average Thickness of Tube .083 in. Area of Section 1.472 sq. in

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg.	Corrected					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
100	100	2430	5100	7500	5100	68.2	44.9	95.0	40.9	0	0	0	0	0	0
300	300	7290	14100	21400	14600	64.1	39.5	89.8	34.9	4.1	5.4	5.2	6.0	5.2	.00026
500	500	12150	23050	35200	23900	59.0	34.7	84.0	30.9	9.2	10.2	11.0	10.0	10.1	.00051
700	700	17000	32100	49100	33300	53.0	29.0	78.9	25.0	15.2	15.9	16.1	15.9	15.8	.00079
900	900	21900	41100	63000	42900	47.3	23.2	74.0	19.4	20.9	21.7	21.0	21.5	21.3	.00107
950	950	23100	43400	66500	45100	46.1	22.0	73.0	18.3	22.1	22.9	22.0	22.6	22.4	.00112
1000	1000	24300	45600	69900	47500	43.9	20.0	71.3	17.0	24.3	24.9	23.7	23.9	24.2	.00121
1050	1050	25500	47900	73400	49700	41.0	17.1	69.3	15.2	27.2	27.8	25.7	25.7	26.6	.00133
1100	1090	26500	50100	76600	52200	38.8	14.1	67.0	13.9	29.4	30.8	28.0	27.0	29.0	.00145
1150	1140	27700	52400	80100	54400	35.4	11.1	65.0	11.8	32.8	33.7	30.0	29.1	31.4	.00157
1200	1190	29200	54500	83500	56800	32.0	7.4	62.5	9.0	36.2	37.5	32.5	31.9	34.5	.00173
1250	1240	30100	56800	86900	59000	26.9	4.0	55.0	4.5	41.3	40.7	40.0	36.4	39.6	.00198
1300	1290	31300	59100	90400	61500	8.0	86.0	47.2	90.0	60.2	58.9	47.8	50.9	54.5	.00273

TABLE XXVIII.

TUBE NO 5

TABLE XXVIII.

TUBE NO 5

Axial Tension Only

Inside Diameter of Tube 5.554 inches

Section AB

Average Thickness of Tube .091 in. Area of Section 1.623 sq. in

Machine Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
		AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
5850	3600	74.0	35.2	72.8	97.7	0	0	0	0	0	0
20600	12700	67.5	27.5	68.0	87.3	6.5	7.7	4.8	10.4	7.3	.00037
30600	18900	63.8	24.5	64.0	81.5	10.2	10.7	8.8	16.2	11.5	.00058
42200	26000	58.0	19.3	58.7	79.0	16.0	15.9	14.1	18.7	16.2	.00081
50100	30900	54.8	16.0	56.0	74.8	19.2	19.2	16.8	22.9	19.5	.00098
61200	37700	47.9	8.2	50.0	69.6	25.1	26.0	21.8	27.1	25.0	.00125
64600	39800	46.9	7.2	49.0	67.8	27.1	28.0	23.8	29.9	27.2	.00136
68100	42000	44.0	5.0	46.0	66.5	30.0	30.2	26.8	31.2	29.5	.00148
72300	44600	41.0	1.1	44.5	64.0	33.0	34.1	28.3	33.7	32.3	.00162
76100	46900	36.6	98.2	43.8	60.8	37.4	37.0	29.0	36.9	35.1	.00176
79800	49200	31.7	94.2	40.3	56.3	42.3	41.0	32.5	41.4	39.3	.00197
82400	50800	26.0	90.7	34.3	49.9	48.0	44.5	38.5	47.8	44.7	.00224
85200	52500	18.5	84.1	26.0	40.5	55.5	51.1	46.8	57.2	52.6	.00263

Section BC

Average Thickness of Tube .092 in. Area of Section 1.641 sq. in.

Machine Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences.				Av. Diff	Av. Unit Elongation
		BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
5850	3600	47.5	54.8	34.7	80.9	0	0	0	0	0	0
20600	12600	38.0	49.5	28.0	73.1	9.5	5.3	6.7	7.8	7.4	.00037
30600	18700	32.5	45.0	25.0	67.8	15.0	9.8	9.7	13.1	11.9	.00060
42200	25800	28.0	39.9	20.0	64.0	19.5	14.9	14.7	16.9	16.5	.00083
50100	30600	25.5	34.9	15.0	60.9	22.0	19.9	19.7	20.0	20.4	.00102
61200	37300	17.8	26.5	9.5	55.1	29.7	28.3	25.2	25.8	27.3	.00137
64600	39400	15.0	25.9	7.1	53.6	32.5	28.8	27.6	27.3	29.1	.00146
68100	41500	14.0	22.1	6.2	52.9	33.5	32.7	28.5	28.0	30.7	.00154
72300	44200	11.0	18.6	3.5	50.1	36.5	36.2	31.2	30.8	33.7	.00169
76100	46400	7.5	16.9	0.8	47.1	40.0	37.9	33.9	33.8	36.4	.00182
79800	48700	2.9	11.0	97.5	44.0	44.6	43.1	37.2	36.9	40.5	.00203
82400	50200	96.5	7.0	93.1	38.0	51.0	47.8	41.6	42.9	45.8	.00229
85200	52000	91.0	1.5	89.0	30.0	56.5	53.3	45.7	50.9	51.6	.00258

Section CD

Average Thickness of Tube .092 in. Area of Section 1.632 sq. in.

Machine Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av Diff.	Av Unit Elongation
		CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
5850	3600	53.0	58.8	40.1	49.0	0	0	0	0	0	0
20600	12700	48.0	52.3	31.1	42.0	5.0	6.5	9.0	7.0	6.9	.00035
30600	18800	40.5	45.5	28.0	37.5	12.5	13.3	12.1	11.5	12.4	.00062
42200	25900	35.0	41.0	26.5	31.0	18.0	17.8	13.6	18.0	16.9	.00085
50100	30700	33.2	36.0	24.1	29.3	19.8	22.8	16.0	19.7	19.6	.00098
61200	37500	29.0	28.2	18.1	23.0	24.0	30.6	22.0	27.0	25.9	.00130
64600	39600	27.0	26.5	16.0	22.0	26.0	32.3	24.1	28.0	27.6	.00138
68100	41800	23.0	21.3	13.2	18.5	30.0	37.5	26.9	31.5	31.5	.00158
72300	44300	19.5	19.0	12.0	18.1	33.5	39.8	28.1	31.9	34.4	.00172
76100	46700	15.6	15.9	9.0	16.1	37.4	42.9	31.1	33.9	36.4	.00182
79800	48900	11.1	9.1	7.0	14.0	41.9	4?.7	33.1	36.0	40.2	.00201
82400	50500	4.3	6.3	2.1	10.6	48.7	?2.6	38.0	39.4	44.7	.00224
85200	52200	97.5	99.5	97.3	8.3	55.5	59.3	42.8	41.7	49.8	.00249

TABLE XXIX.

TUBE NO. 6.

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.579 inches

Axial Gage Lines

TABLE XXIX.

TUBE NO. 6.

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.579 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .083 in. Area of Section 1.467 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff	Av. Unit Elongation
Gage Rdg	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	95	2300	3400	5700	3900	6.9	0.9	8.0	65.2	0	0	0	0	0	0
300	[illegible]	7000	9000	16000	10900	3.0	96.5	5.5	61.2	3.9	4.4	2.5	1.0	3.5	.00018
500	[illegible]	11600	14600	27200	17900	99.9	92.5	1.3	58.0	7.0	8.4	6.7	7.2	7.3	.00037
600	[illegible]	4100	17400	31500	21500	98.0	90.0	99.8	56.5	8.9	10.9	8.2	8.7	9.2	.00046
700	[illegible]	16400	20200	36600	24900	96.2	87.5	97.9	54.0	10.7	13.4	10.1	11.2	11.4	.00057
750	720	17600	21600	39200	26700	94.9	86.0	95.8	52.6	12.0	14.9	12.2	12.6	12.9	.00065
800	765	18700	23000	41700	28400	94.0	85.0	94.5	51.0	12.9	15.9	13.5	14.2	14.1	.00071
850	815	19900	24400	44300	30200	92.0	83.5	92.5	49.1	14.9	17.4	15.5	16.1	16.0	.00080
900	865	21200	25800	47000	32000	89.1	81.5	89.6	44.9	17.8	19.4	18.4	20.3	19.0	.00095
950	915	22400	27200	49600	33800	71.0	68.0	68.5	25.0	35.0	32.9	39.5	40.2	36.9	.00185

Section BC

Average Thickness of Tube .083 in. Area of Section 1.467 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Av. Diff	Av. Unit Elongation
Gage Rdg.	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
100	95	2300	3400	5700	3900	46.0	75.9	88.0	66.6	0	0	0	0	0	0
300	285	7000	9000	16000	10900	43.0	71.0	84.0	63.1	3.0	4.9	4.0	3.5	3.9	.00020
500	475	11600	14600	27200	17900	40.0	68.0	80.0	60.1	6.0	7.9	8.0	6.5	7.1	.00036
600	575	14100	17400	31500	21500	38.0	67.0	78.0	58.0	8.0	8.9	10.0	8.6	8.9	.00045
700	670	16400	20200	36600	24900	36.0	65.0	75.0	56.0	10.0	10.9	13.0	10.6	11.1	.00056
750	720	17600	21600	39200	26700	35.0	63.5	73.3	53.7	11.0	12.4	14.7	12.9	12.8	.00064
800	765	18700	23000	41700	28400	34.0	62.0	72.0	52.5	12.0	13.9	16.0	14.1	14.0	.00070
850	815	19900	24400	44300	30200	32.0	59.5	69.0	50.5	14.0	16.4	19.0	16.1	16.4	.00082
900	865	21200	25800	47000	32000	29.8	58.7	65.0	47.5	16.2	17.2	23.0	19.1	18.9	.00095
950	915	22400	27200	49600	33800	10.0	44.0	44.0	24.5	36.0	31.9	44.0	42.1	38.5	.00193

Section CD

Average Thickness of Tube .080 in. Area of Section 1.422 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg	Corrected					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
[illegible]	95	2300	3400	5700	4000	73.4	54.0	10.9	94.8	0.	0	0	0	0	0
300	285	7000	9000	16000	11200	68.9	48.0	7.2	90.9	4.5	6.0	3.7	3.9	4.5	.00023
[illegible]	[illegible]	11600	14600	27200	18400	65.6	45.1	2.3	86.7	7.8	8.9	8.6	8.2	8.4	.00042
600	[illegible]	14100	17400	31500	22100	63.3	43.6	0.4	85.0	10.1	10.4	10.5	9.8	10.2	.00051
[illegible]	[illegible]	16400	20200	36600	25700	61.1	40.9	98.1	82.0	12.3	13.1	12.8	12.8	12.3	.00062
750	[illegible]	17600	21600	39200	27600	60.0	39.1	96.9	80.2	13.4	14.9	14.0	14.6	14.2	.00071
[illegible]	[illegible]	18700	23000	41700	29300	59.1	37.0	94.7	79.1	14.3	17.0	16.2	15.7	15.8	.00079
[illegible]	[illegible]	[illegible]	24400	[illegible]	31100	58.0	34.1	93.1	77.0	15.4	19.9	17.8	17.8	18.0	.00090
[illegible]	[illegible]	[illegible]	25800	47000	33000	53.4	29.0	85.4	74.0	20.0	25.0	25.5	20.8	22.8	.00114
[illegible]	[illegible]	[illegible]	27200	49600	34900	24.0	0.0	58.0	49.0	49.4	54.0	52.9	45.8	50.5	.00253

TABLE XXX.

TUBE NO. 7

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.588 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .091 in Area of Section 1.623 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	95	2330	7950	10300	6300	31.0	82.3	40.3	79.3	0	0	0	0	0	.0
200	190	4700	15300	20000	12300	26.7	79.0	36.0	74.7	4.3	3.3	4.3	4.6	4.1	.00021

TABLE XXX.

TUBE NO. 7

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.588 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .091 in Area of Section 1.623 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av Axial Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	95	2330	7950	10300	6300	31.0	82.3	40.3	79.3	0	0	0	0	0	.0
200	190	4700	15300	20000	12300	26.7	79.0	36.0	74.7	4.3	3.3	4.3	4.6	4.1	.00021
300	285	7000	22650	29700	17300	23.9	75.0	32.1	69.9	7.1	7.3	8.2	9.4	8.0	.00040
400	380	9300	30000	39300	24200	18.0	67.9	29.0	65.9	13.0	14.4	11.3	13.4	13.0	.00065
450	430	10500	33600	44100	27200	14.9	62.6	26.0	64.3	16.1	19.7	14.3	15.0	16.3	.00082
500	475	11600	37300	48900	30100	10.0	53.5	14.5	60.0	21.0	28.8	25.8	19.3	23.7	.00119
550	525	12900	41000	53900	33100	5.7	47.8	8.0	53.3	25.3	34.5	32.3	26.0	29.5	.00148

Section BC

Average Thickness of Tube .091 in. Area of Section 1.623 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Av. Diff	Av. Unit Elongation
Gage Rdg	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
100	95	2330	7950	10300	6300	39.0	35.0	54.0	32.0	0	0.	0	0	0	0
200	190	4700	15300	20000	12300	35.0	30.3	52.1	28.0	4.0	4.7	1.9	4.0	3.7	.00019
300	285	7000	22650	29700	17300	31.1	26.0	47.0	24.0	7.9	9.0	7.0	8.0	8.0	.00040
400	380	9300	30000	39300	24200	28.0	19.8	42.5	19.8	11.0	15.2	11.5	12.2	12.5	.00063
450	430	10500	33600	44100	27200	25.1	15.0	39.0	19.2	13.9	20.0	15.0	12.8	15.4	.00077
500	475	11600	37300	48900	30100	20.0	8.0	31.5	16.3	19.0	27.0	22.5	15.7	21.1	.00106
550	525	12900	41000	53900	33100	12.4	1.3	24.0	12.3	26.6	33.7	30.0	19.7	27.5	.00138

Section CD

Average Thickness of Tube .C91 in. Area of Section 1.623 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Gage Rdg	Corrected					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
100	95	2330	7950	10300	6300	46.0	72.0	66.9	48.2	0	0	0	0	0	0
200	190	4700	15300	20000	12300	42.1	67.3	62.2	45.6	3.9	4.7	4.7	2.6	4.0	.00020
300	285	7000	22650	29700	17300	38.0	63.3	58.9	41.0	8.0	8.7	8.0	7.2	8.0	.00040
400	380	9300	30000	39300	24200	33.0	59.7	54.0	37.2	13.0	12.3	12.9	11.0	12.3	.00062
450	430	10500	33600	44100	27200	28.6	55.6	51.1	34.0	17.4	16.4	15.8	14.2	16.0	.00080
500	475	11600	37300	48900	30100	21.2	52.5	46.7	27.1	24.8	19.5	20.2	21.1	21.4	.00107
550	525	12900	41000	53900	33100	13.8	40.0	39.0	17.0	32.2	32.0	27.9	31.2	30.8	.00154

TABLE XXXI.

TUBE NO 8.

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.560 inches

Axial Gage Lines

Section A B

Average Thickness of Tube .094 in Area of Section 1.688 sq. in

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Readings on Gage Line				Differences				Av. Diff.	Av. Unit Elongation
Guge Rdg.	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	100	2450	3400	5850	3500	48.9	1.1	2.9	16.9	0	0	0	0	0	0
400	400	9800	11800	21600	12800	43.7	95.9	97.5	12.6	5.2	5.2	5.4	4.3	5.0	.00025
700	700	17100	20200	37300	22100	38.1	92.0	92.7	6.0	10.8	9.1	10.2	10.9	10.2	.00051
1000	1000	24500	28600	53100	31400	27.9	87.0	84.5	93.5	21.0	14.1	18.4	23.4	19.2	.00096
1100	1100	26500	31400	57900	34400	21.0	81.5	77.5	86.4	27.9	19.6	25.4	30.5	25.9	.00130

TABLE XXXI.

TUBE NO 8.

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.560 inches

Axial Gage Lines

Section AB

Average Thickness of Tube .094 in Area of Section 1.688 sq.in

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs	Av. Axial Unit Stress lbs. sq. in	Readings on Gage Line				Differences				Av. Diff.	Av Unit Elongation
Gage Rdg.	Corrected					AB1	AB2	AB3	AB4	AB1	AB2	AB3	AB4		
100	100	2450	3400	5850	3500	48.9	1.1	2.9	16.9	0	0	0	0	0	0
400	400	9800	11800	21600	12800	43.7	95.9	97.5	12.6	5.2	5.2	5.4	4.3	5.0	.00025
700	700	17100	20200	37300	22100	38.1	92.0	92.7	6.0	10.8	9.1	10.2	10.9	10.2	[illegible]
1000	1000	24500	28600	53100	31400	27.9	87.0	84.5	93.5	21.0	14.1	18.4	23.4	19.2	[illegible]
1100	1100	26500	31400	57900	34400	21.0	81.5	77.5	86.4	27.9	19.6	25.4	30.5	25.9	[illegible]
1200	1200	28900	34200	63100	37400	5.5	71.0	61.0	65.0	43.4	30.1	41.9	51.9	41.8	.00209

Section BC

Average Thickness of Tube .094 in. Area of Section 1.688 sq in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in	Readings on Gage Line				Differences				Av. Diff	Av. Unit Elongation
Gage Rdg.	Corrected					BC1	BC2	BC3	BC4	BC1	BC2	BC3	BC4		
[illegible]	[illegible]	2450	3400	5850	3500	64.9	64.1	59.9	50.4	0	0	0	0	0	0
[illegible]	400	9800	11800	21600	12800	60.9	59.9	55.9	44.5	4.0	4.2	4.0	5.9	4.5	[illegible]
700	700	17100	20200	37300	22100	55.1	54.7	52.0	37.6	9.8	9.4	7.9	12.8	10.0	[illegible]
1000	1000	24500	28600	53100	31400	42.5	50.5	44.3	22.0	22.4	13.6	15.6	28.4	20.0	[illegible]
[illegible]	1100	26500	31400	57900	34400	33.0	49.5	42.0	13.5	31.9	14.6	17.6	36.9	25.2	[illegible]
[illegible]	1200	28900	34200	63100	37400	18.0	31.5	30.5	98.0	46.9	32.6	29.4	52.4	40.3	[illegible]

Section CD

Average Thickness of Tube .094 in. Area of Section 1.688 sq. in.

Water Pressure		Axial Load due to W. Pressure	Machine Load lbs.	Total Axial Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Readings on Gage Line				Differences.				Av. Diff	Av. Unit Elongation
Gag. Rdg.	Corrected					CD1	CD2	CD3	CD4	CD1	CD2	CD3	CD4		
100	100	2450	3400	5850	3500	65.1	86.1	64.9	71.2	0	0	0	0	0	0
400	400	9800	11800	21600	12800	60.0	81.4	60.0	66.1	5.1	4.7	4.9	5.1	5.0	.00025
700	700	17100	20200	37300	22100	53.9	77.3	54.9	60.2	11.2	8.8	10.0	11.0	10.2	.00051
1000	1000	24500	28600	53100	31400	49.2	72.0	45.3	47.2	15.9	14.1	19.6	24.0	18.9	.00095
1100	1100	26500	31400	57900	34400	45.4	61.5	35.3	38.9	19.7	24.6	29.6	31.3	26.3	.00132
1200	1200	28900	34200	63100	37400	15.0	29.5	1.0	20.6	50.1	56.6	63.9	50.6	55.3	.00277

TABLE XXXII.

TUBE NO. 1.

Ratio of Hoop to Axial Tension 0.24
Inside Diameter of Tube 5.564 inches.
Circumferential Gage Lines
A Lines

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
50	46	10200	1600	35.7	14.1	35.9	65.3	0	0	0	0	0	0
150	145	29600	4600	36.1	15.0	36.0	66.6	- 0.4	- 0.9	- 0.1	-1.3	- 0.7	-.00004
225	215	44000	6900	36.1	16.9	37.0	67.1	- 0.4	-2.8	- 1.1	-1.8	- 1.5	-.00008
275	263	53800	8400	37.0	18.0	38.1	67.0	- 1.3	-3.9	- 2.2	-1.7	- 2.3	-.00012
300	285	58500	9200	37.2	18.3	39.0	67.1	- 1.5	-4.2	- 3.1	-1.8	- 2.7	-.00014
325	310	63400	9900	37.9	19.1	39.8	67.0	- 22	-5.0	- 3.9	-1.7	- 3.2	-.00016
350	335	68200	10600	38.8	19.9	40.1	66.7	- 3.1	-5.8	- 4.2	-1.4	- 3.6	-.00018
375	357	73100	11400	39.0	19.5	40.9	66.0	- 3.3	- 5.4	- 5.0	-07	- 3.6	-.00018
400	380	77900	12300	40.0	20.1	42.0	64.8	-4.3	-6 0	- 6.1	-0.5	- 40	-.00020
425	405	82800	13000	41.2	20.9	43.0	64.0	-5.5	-6.8	- 7.1	-1.3	- 4.5	-.00023
450	430	87700	13700	43.0	21.0	44.0	63.1	-7.3	-6.9	- 8.1	-2.2	-5.0	-.00025
475	453	92400	14500	43.9	22.9	43.0	64.1	-8.2	- 8.8	- 7.1	-1.2	-5.7	-.00029
500	475	97100	15200	45.0	25.8	41.8	70.1	-93	- 117	-5.9	-4.8	- 7.9	-.00040

TABLE XXXII.

TUBE NO. 1.

Ratio of Hoop to Axial Tension 0.24

Inside Diameter of Tube 5.564 inches.

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs	Average Unit Stress lbs sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
50	46	10200	1500	80.8	39.0	50.2	47.5	0	0	0	0	0	0
150	145	29600	4500	81.8	40.1	50.8	48.8	-1.0	-1.1	-0.6	-.8	1.0	.00005
225	215	44000	6600	82.6	40.8	51.0	49.6	-1.8	-1.8	-0.8	2.1	1.6	.00008
275	263	53800	8200	83.4	41.8	51.2	50.1	-2.6	-2.8	-1.0	2.6	2.3	.00012
300	285	58500	8800	84.0	41.8	52.0	50.0	-3.2	-2.8	1.8	2.5	2.6	.00013
325	310	63400	9600	84.1	42.0	52.1	50.1	-3.3	-3.0	-1.9	-2.6	2.7	.00014
350	335	68200	10300	84.0	43.0	52.0	50.2	-3.2	4.0	-1.8	-2.7	2.9	-.00015
375	357	73100	11000	84.4	43.0	52.2	50.8	-3.6	-4.0	-2.0	-3.3	3.2	-.00016
400	380	77900	11800	85.0	43.0	52.5	50.9	-4.2	-4.0	2.3	3.4	3.5	-.00018
425	405	82800	12500	85.8	43.8	52.6	51.7	-5.0	-4.8	-2.4	4.2	4.1	.00021
450	430	87700	13200	85.0	43.0	53.0	52.8	-4.2	-4.0	2.8	5.3	-4.1	-.00021
475	453	92400	14000	85.5	42.0	52.7	52.0	4.7	-3.0	-2.5	4.5	-3.7	.00019
500	475	97100	14700	86.9	41.4	53.1	52.5	-6.1	-2.4	-2.9	5.0	4.1	.00021

B Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B		
50	46	10200	1500	40.0	37.1	31.9	62.6	0	0	0	0	0	0
150	145	29600	4500	41.1	38.2	32.1	63.8	1.1	-1.1	-0.2	1.2	0.9	-.00005
225	215	44000	6600	41.9	39.0	32.0	64.0	-1.9	-1.9	-0.1	-1.4	1.3	-.00007
275	263	53800	8200	42.7	39.1	33.0	65.0	-2.7	-2.0	-1.1	-2.4	-2.1	-.00011
300	285	58500	8800	43.6	39.8	33.1	65.1	-3.6	-2.7	-1.2	-2.5	-2.5	-.00013
325	310	63400	9600	43.9	39.7	33.8	65.1	-3.9	-2.6	-1.9	-2.5	-2.7	-.00014
350	335	68200	10300	44.7	40.0	34.1	65.1	-4.7	-2.9	-2.2	-2.5	-3.1	-.00016
375	357	73100	11000	45.9	39.1	34.3	64.9	-5.9	-2.0	-2.4	-2.3	-3.2	-.00016
400	380	77900	11800	46.4	39.0	34.9	65.0	-6.4	-1.9	-3.0	-2.4	-3.4	.00017
425	405	82800	12500	47.0	39.0	35.7	65.0	7.0	-1.9	-3.8	2.4	-3.8	.00019
450	430	87700	13200	48.2	38.1	36.8	65.0	-8.2	-1.0	-4.9	2.4	4.1	-.00021
475	453	92400	14000	48.8	37.5	37.0	64.7	-8.8	-0.4	-5.1	-2.1	-4.1	-.00021
500	475	97100	14800	50.0	37.9	37.5	64.0	-10.0	-0.8	-5.6	-1.4	-4.5	-.00023

C Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C		
50	46	10200	1600	12.8	46.0	23.3	28.1	0	0	0	0	0	0
150	145	29600	4500	13.0	47.7	24.0	29.4	-0.2	-1.7	-0.7	-1.3	-1.0	-.00005
225	215	44000	5700	14.0	48.8	24.2	30.2	-1.2	-2.8	-0.9	-2.1	-1.8	-.00009
275	263	53800	8300	15.0	49.1	25.1	30.9	-2.2	-3.1	-1.8	-2.8	-2.5	-.00013
300	285	58500	9000	15.8	49.6	25.7	31.2	-3.0	-3.6	-2.4	-3.1	-3.0	-.00015
325	310	63400	9700	17.0	49.1	26.5	31.1	-4.2	-3.1	-3.2	-3.0	-3.4	-.00017
350	335	68200	10500	17.9	49.1	27.0	31.2	-5.1	-3.1	-3.7	-3.1	-3.8	-.00019
375	357	73100	11200	19.0	48.3	27.7	29.7	-6.2	-2.3	-4.4	-1.5	-3.6	-.00018
400	380	77900	12100	20.9	48.0	29.0	28.9	-8.1	-2.0	-5.7	0.8	-4.2	-.00021
425	405	82800	12800	23.0	47.7	29.8	28.2	-10.2	-1.7	-6.5	-0.1	-4.6	-.00023
450	430	87700	13500	25.0	47.5	31.5	27.9	-12.2	-1.5	-8.2	-0.2	-5.4	-.00027
475	453	92400	14300	26.8	47.0	32.0	27.0	-14.0	-1.0	-8.7	-1.1	-5.1	-.00029
500	475	97100	15000	27.2	48.7	32.8	27.8	-14.4	-2.7	-9.5	-0.3	-6.6	-.00033

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
50	46	10200	1600	35.7	14.1	35.9	65.3	0	0	0	0	0	0
150	145	29600	4600	36.1	15.0	36.0	66.6	-0.4	-0.9	-0.1	-1.3	-0.1	-.00004
225	215	44000	6900	36.1	16.9	37.0	67.1	-0.4	-2.8	-1	-1.8	-1.5	-.00008
275	263	53800	8400	37.0	18.0	38.1	67.0	-1.3	-3.9	-2.2	-1.7	-2.3	-.00012
300	285	58500	9200	37.2	18.3	39.0	67.1	-1.5	-4.2	-3.1	-1.8	2.7	-.00014
325	310	63400	9900	37.9	19.1	39.8	67.0	-2.2	-5.0	-3.9	-1.7	-3.2	-.00016
350	335	68200	10600	38.8	19.9	40.1	66.7	-3.1	-5.8	-4.2	-1.4	-3.6	-.00018
375	357	73100	11400	39.0	19.5	40.9	66.0	-3.3	-5.4	-5.0	-0.7	-3.6	-.00018
400	380	77900	12300	40.0	20.1	42.0	64.8	-4.3	-6.0	-6.1	-0.5	-4.0	-.00020
425	405	82800	13000	41.2	20.9	43.0	64.0	-5.5	-6.8	-7.1	-1.3	-4.5	-.00023
450	430	87700	13700	43.0	21.0	44.0	63.1	-7.3	-6.9	-8.1	-2.2	-5.0	-.00025
475	453	92400	14500	43.9	22.9	43.0	64.1	-8.2	-8.8	-7.1	-1.2	-5.7	.00029
500	475	97100	15200	45.0	25.8	41.8	70.1	-9.3	-11.7	-5.9	-4.8	-7.9	-.00040

TABLE XXXIII.

TUBE NO 2

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.558 inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average Unit
Gage Rdg.	Corrected	lbs	lbs. sq. in.	1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A	Difference	Elongation
100	95	10200	3100	94.9	67.9	45.4	51.5	0	0	0	0	0	0
300	285	29300	9000	93.0	67.3	43.7	50.9	1.9	0.6	1.7	0.6	1.2	.00006
500	475	48500	15000	91.0	67.3	43.5	51.1	3.9	0.6	1.9	0.4	1.7	.00008
600	575	58300	18000	89.9	66.8	43.0	50.7	5.0	1.1	2.4	0.8	2.3	.00012

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average Unit
Gage Rdg.	Corrected	lbs.	lbs. sq. in	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Elongation
100	95	10200	3100	17.0	48.1	82.1	64.9	0	0	0	0	0	0
300	285	29300	8900	16.4	47.0	80.0	64.1	0.6	1.1	2.1	0.8	1.2	.00006
500	475	48500	14600	16.1	46.0	78.0	63.9	0.9	2.1	4.1	1.0	2.0	.00010
600	575	58300	17700	15.2	46.4	76.7	62.2	1.8	1.7	5.4	2.7	2.9	.00015
700	670	67900	20600	13.9	46.3	76.1	63.0	3.1	1.8	6.0	1.9	3.2	.00016
800	765	77200	23400	12.5	45.2	75.1	62.9	4.5	2.9	7.0	2.0	4.1	.00021
850	815	82200	24900	11.9	45.2	74.5	61.9	5.1	2.9	7.6	3.0	4.7	.00024
900	865	87100	26400	10.8	45.0	74.9	61.0	6.2	3.1	7.2	3.9	5.1	.00026
950	915	92000	27800	10.3	46.2	74.4	61.3	6.7	1.9	7.7	3.6	5.0	.00025
1000	965	96800	29300	7.1	47.9	73.7	62.2	9.9	0.2	8.4	2.7	5.3	.00027

TABLE XXXIII.

TUBE NO 2

Ratio of Hoop to Axial Tension 0.475
Inside Diameter of Tube 5.558 inches
Circumferential Gage Lines

A Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs	lbs sq. in.	1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A	Difference	Unit Elongation
100	95	10200	3100	94.9	67.9	45.4	51.5	0	0	0	0	0	0
300	285	29300	9000	93.0	67.3	43.7	50.9	1.9	0.6	1.7	[illegible]	1.2	.00006
500	475	48500	15000	91.0	67.3	43.5	51.1	3.9	0.6	1.9	0.4	[illegible]	[illegible]
600	575	58300	18000	89.9	66.8	43.0	50.7	5.0	1.1	2.4	0.8	2.3	00012
700	670	67900	20900	90.0	67.1	42.2	51.6	4.9	0.8	3.2	-0.1	[illegible]	.00011
800	765	77200	23800	89.5	67.6	41.4	51.7	5.4	0.3	4.0	-0.2	2.4	00012
850	815	82200	25300	88.0	66.2	41.0	51.3	6.9	1.7	4.4	0.2	3.3	.00017
900	865	87100	26800	87.0	65.7	42.0	52.1	7.9	2.2	3.4	-0.6	3.2	[illegible]
950	915	92000	28300	86.0	66.0	39.5	51.2	8.9	1.9	5.9	0.3	4.3	[illegible]
1000	965	96800	29800	86.4	68.1	40.2	53.2	8.5	0.2	5.2	-1.7	3.0	00015

B Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs	lbs sq. in	1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B	Difference	Unit Elongation
100	95	10200	3100	57.0	74.5	13.0	49.5	0	0	0	0	0	0
300	285	29300	9000	56.1	73.9	4.9	49.2	0.9	0.6	-1.9	0.3	-0.1	-.00001
500	475	48500	15000	54.0	74.3	12.0	49.0	3.0	0.2	1.0	0.5	1.2	.00006
600	575	58300	18000	53.4	74.0	9.5	49.6	3.6	0.5	3.5	[illegible]	1.9	.00010
700	670	67900	20900	52.1	72.7	7.1	51.1	4.9	1.8	5.9	-1.6	2.8	00014
800	765	77200	23800	53.1	71.9	6.8	52.0	3.9	2.6	6.2	[illegible]	2.5	00014
850	815	82200	25300	51.9	70.8	3.9	49.9	5.1	3.7	9.1	-0.4	4.4	.00022
900	865	87100	26800	53.5	70.1	3.8	51.8	3.5	4.4	9.2	-2.3	3.7	.00019
950	915	92000	28300	51.3	71.0	3.3	51.0	5.7	3.5	9.7	-1.5	4.4	.00022
1000	965	96800	29800	48.9	75.6	2.0	54.5	8.1	-1.1	11.0	-5.0	3.3	.00017

C Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs	lbs. sq. in.	1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C	Difference	Unit Elongation
100	95	10200	3100	34.0	58.8	13.9	35.3	0	0	0	0	0	0
300	285	29300	9000	33.3	57.3	11.1	34.0	0.7	1.5	2.8	1.3	1.6	.00008
500	475	48500	14800	32.1	57.8	9.0	34.4	1.9	1.0	4.9	0.9	2.2	.00011
600	575	58300	17800	31.3	56.5	8.0	34.0	2.7	2.3	5.9	1.3	3.1	00016
700	670	67900	20800	30.9	56.2	5.9	33.4	3.1	2.6	8.0	1.9	3.9	.00020
800	765	77200	23600	30.1	56.9	4.8	32.9	3.9	1.9	9.1	2.4	4.3	00022
850	815	82200	25100	28.2	55.9	2.8	31.0	5.8	2.9	11.1	4.3	6.0	00030
900	865	87100	26600	28.9	57.4	1.5	31.1	5.1	1.4	12.4	4.2	5.8	00029
950	915	92000	28100	28.1	58.0	0.9	32.0	5.9	0.8	13.0	3.3	5.8	.00029
1000	965	96800	29600	26.9	59.3	98.5	34.5	7.1	-0.5	15.4	0.8	5.7	.00029

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs.	lbs. sq. in	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Unit Elongation
100	95	10200	3100	17.0	48.1	82.1	64.9	0	0	0	0	0	0
300	285	29300	8900	16.4	47.0	80.0	64.1	0.6	1.1	2.1	0.8	1.2	00006
500	475	48500	14600	16.1	46.0	78.0	63.9	0.9	2.1	4.1	[illegible]	2.0	00010
600	575	58300	17700	15.2	46.4	76.7	62.2	1.8	1.7	5.4	2.7	2.9	00015
700	670	67900	20600	13.9	46.3	76.1	63.0	3.1	1.8	6.0	1.9	3.2	00016
800	765	77200	23400	12.5	45.2	75.1	62.9	4.5	2.9	7.0	2.0	4.1	00021
850	815	82200	24900	11.9	45.2	74.5	61.9	5.1	2.9	7.6	3.0	4.7	00024
900	865	87100	26400	10.8	45.0	74.9	61.0	6.2	3.1	7.2	3.9	5.1	00026
950	915	92000	27800	10.3	46.2	74.4	61.3	6.7	1.9	7.7	3.6	5.0	00025
1000	965	96800	29300	7.1	47.9	73.7	62.2	9.9	0.2	8.4	2.7	5.3	00027

TABLE XXXIV.

TUBE NO 3

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.561 Inches.

Circumferential Gage Lines

A Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs.	lbs. sq. in	1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A	Difference	Unit Elongation
100	100	6025	3600	64.4	54.9	83.8	23.0	0	0	0	0	0	0
400	400	21600	12800	59.4	50.0	79.0	20.3	5.0	4.9	4.8	2.7	4.5	.00023
700	700	37200	22100	55.4	47.0	73.0	16.0	9.0	7.9	10.8	7.0	8.7	.00044
900	900	47800	28300	53.0	45.2	68.0	13.0	11.4	9.7	15.8	10.0	11.7	.00059
1100	1090	57800	34200	52.9	43.9	66.4	12.5	11.5	11.0	17.4	10.5	12.6	.00063

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs.	lbs. sq. in	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Unit Elongation
100	100	6025	3600	57.5	26.8	56.0	34.5	0	0	0	0	0	0
400	400	21600	12900	51.0	25.1	51.1	33.1	6.5	1.7	4.9	1.4	3.6	.00018
700	700	37200	22200	44.5	21.2	46.2	28.4	13.0	5.6	9.8	6.1	8.5	.00043
900	900	47800	28500	41.9	19.2	44.9	28.0	15.6	7.6	11.1	6.5	10.2	.00051
1100	1090	57800	34700	38.9	17.9	43.1	26.0	18.6	8.9	12.9	8.5	12.2	.00061
1300	1290	68300	40900	35.9	15.9	39.5	23.3	21.6	10.9	16.5	11.2	15.1	.00076
1400	1390	73500	44100	34.1	14.9	38.8	22.5	23.4	11.9	17.2	12.0	16.1	.00081
1600	1590	84000	50400	29.0	11.0	37.3	20.9	28.5	15.8	18.7	13.6	19.2	.00096
1700	1690	89200	53500	22.0	6.0	31.0	11.9	35.5	20.8	25.0	22.6	26.0	.00130
1750	1740	91400	54800	8.0	95.0	2.0	0.0	49.5	31.8	54.0	34.5	42.5	.00213

TABLE XXXIV.

TUBE NO 3

Ratio of Hoop to Axial Tension 0.92

Inside Diameter of Tube 5.561 Inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
100	100	6025	3600	644	549	83.8	23.0	0	0	0	0	0	0
400	400	21600	12800	594	500	790	203	5.0	4.9	48	2.7	45	00023
700	700	37200	22100	554	47.0	730	160	9.0	79	108	7.0	87	.00044
900	900	47800	28300	530	452	680	13.0	11.4	97	158	100	11.7	00059
1100	1090	57800	34200	529	439	664	125	11.5	11 0	174	105	126	.00063
1300	1290	68300	40500	509	410	645	11.7	135	13.9	193	11.3	14.5	.00073
1400	1390	73500	43600	490	40.1	631	103	154	14.8	207	127	159	.00080
1600	1590	84000	49800	490	360	604	98	154	189	234	132	177	.00089
1700	1690	89200	52900	410	300	570	4.0	234	249	268	190	235	00118
1750	1740	91400	54200	12.0	17.5	470	971	524	374	368	259	381	.00191

B Lines

Water Pressure		Total Load lbs	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B		
100	100	6025	3600	3.0	619	900	444	0	0	0	0	0	0
400	400	21600	12800	99.4	60.0	85.0	422	36	19	50	22	32	00016
700	700	37200	22100	94.0	56.6	79.6	391	90	5.3	10.4	53	7.5	.00038
900	900	47800	28400	91.9	55.7	75.0	37.6	11.1	6.2	15.0	68	9.8	00049
1100	1090	57800	34300	88.0	53.8	740	35.9	15.0	8.1	16.0	85	11.9	.00060
1300	1290	68300	40600	859	51.9	71.8	33.3	17.1	10.0	18.2	11.1	14.1	00071
1400	1390	73500	43700	84.8	51.0	70.0	32.4	18.2	10.9	20.0	12.0	15.3	.00077
1600	1590	84000	50000	79.8	49.0	66.9	28.4	23.2	12.9	23.1	160	18.8	.00094
1700	1690	89200	53000	75.0	45.0	61.0	25.1	28.0	16.9	29.0	19.3	23.3	.00117
1750	1740	91400	54400	57.0	31.0	48.0	14.0	43.0	30.9	42.0	30.4	36.6	.00183

C Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C		
100	100	6025	3600	94.1	69.0	63.0	349	0	0	0	0	0	0
400	400	21600	12900	880	64.5	57.9	328	61	45	51	21	45	.00023
700	700	37200	22200	82.5	62.0	52.0	290	116	70	110	5.9	89	.00045
900	900	47800	28400	790	609	50.1	280	151	81	129	69	108	.00054
1100	1090	57800	34500	76.5	60.0	49.0	260	176	90	140	8.9	12.4	.00062
1300	1290	68300	40700	733	58.2	45.3	235	20.8	10.8	17.7	11.4	15.2	00076
1400	1390	73500	43900	71.1	57.2	45.0	230	23.0	118	180	119	16.2	00081
1600	1590	84000	50200	66.3	54.8	42.9	278	27.8	142	20.1	149	192	.00096
1700	1690	89200	53200	57.2	50.1	39.1	369	36.9	189	239	178	244	.00122
1750	1740	91400	54600	30.0	5.0	33.9	641	64.1	640	291	249	455	.00228

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
100	100	6025	3600	575	26.8	56.0	345	0	0	0	0	0	0
400	400	21600	12900	51 0	25.1	51.1	33.1	65	1.7	49	14	36	00018
700	700	37200	22200	445	21.2	462	28.4	13.0	5.6	98	61	8[illegible]	.00043
900	900	47800	28500	419	19.2	449	28.0	15.6	7.6	11.1	6.5	10.2	00051
1100	1090	57800	34700	389	17.9	43.1	260	18.6	89	129	8.5	122	00061
1300	1290	68300	40900	359	15.9	395	233	216	10.9	165	11.2	151	.00076
1400	1390	73500	44100	34.1	149	38.8	22.5	234	119	17.2	120	161	.00081
1600	1590	84000	50400	290	110	373	20.9	28.5	15.8	187	136	192	00096
1700	1690	89200	53500	220	6.0	31.0	11.9	35.5	208	250	226	260	00130
1750	1740	91400	54800	8.0	95.0	2.0	0.0	49.5	318	540	345	425	00213

TABLE XXXV

TUBE NO. 4

Ratio of Hoop to Axial Tension 0.69

Inside Diameter of Tube 5.563 inches

Circumferential Gage Lines

A Lines

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg.	Corrected	lbs	lbs. sq.in.	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Unit Elongation
100	100	7500	3500	78.0	970	91.1	67.0	0	0	0	0	0	0
300	300	21400	10200	78.0	947	912	63.1	0	2.3	[illegible]	39	15	.00008
500	500	35200	16700	758	940	890	601	2.2	[illegible]	[illegible]	6.9	36	.00018
700	700	49100	23100	73.0	909	88.0	571	50	61	3.1	95	60	.00030
900	900	63000	29800	712	89.5	86.2	548	68	7.5	49	122	79	.00040
950	950	66500	31300	710	898	858	543	70	7.2	53	12.7	[illegible]	.00041
1000	1000	69900	32900	689	87.8	840	539	91	9.2	7.1	131	9.6	.00048
1050	1050	73400	34600	690	881	850	53.9	90	8.9	6.1	13.1	93	.00047
1100	1090	76600	36300	690	890	845	524	90	80	6.5	146	9.5	.00048
1150	1140	80100	37800	681	887	839	521	99	83	72	149	10.1	.00051
1200	1190	83500	39500	680	880	840	510	100	9.0	7.1	160	10.5	.00053
1250	1240	86900	40900	678	87.5	830	511	102	95	8.1	15.9	109	.00055
1300	1290	90400	42700	650	899	79.0	51.1	130	75	121	159	120	.00060

TABLE XXXV

TUBE NO. 4

Ratio of Hoop to Axial Tension 0.69

Inside Diameter of Tube 5563 inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
100	100	7500	3400	19	249	33	54	0	0	0	0	0	0
300	300	21400	9800	91	211	50	40	12	38	-17	4	0[illegible]	.00003
500	500	35200	16200	90	180	50	08	11	69	-17	46	2[illegible]	00011
700	700	49100	22500	89	150	34	980	[illegible]	99	-01	74	4	00021
900	900	63000	28900	70	140	12	970	19	109	21	84	56	00028
950	950	66500	30400	70	130	10	960	29	119	23	94	6[illegible]	00031
1000	1000	69900	32100	60	130	08	954	19	119	25	[illegible]	6[illegible]	00033
1050	1050	73400	33600	55	130	10	960	24	119	23	94	6[illegible]	00033
1100	1090	76600	35400	59	129	09	960	20	[illegible]	24	94	65	00033
1150	1140	80100	37000	49	125	01	952	30	124	[illegible]	102	72	000[illegible]
1200	1190	83500	38500	46	110	00	950	33	13[illegible]	33	104	7[illegible]	000[illegible]9
1250	1240	86900	40000	36	107	999	949	43	142	3[illegible]	105	8[illegible]	0004[illegible]
1300	1290	90400	41600	29	99	992	929	50	150	41	25	92	[illegible]046

B Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B		
100	100	7500	3500	499	909	939	00	0	0	0	0	0	0
300	300	21400	9900	505	880	930	979	-06	29	09	21	13	00007
500	500	35200	16400	498	850	925	950	01	59	14	50	31	00016
700	700	49100	22800	480	835	919	931	19	74	20	69	46	.00023
900	900	63000	29300	460	810	880	920	39	99	59	80	69	00035
950	950	66500	30800	460	811	875	910	39	98	64	90	73	00037
1000	1000	69900	32500	451	799	85[illegible]	900	48	110	80	100	85	00043
1050	1050	73400	34100	450	800	862	902	49	109	77	98	84	00042
1100	1090	76600	35800	443	790	850	900	56	119	89	100	9[illegible]	00046
1150	1140	80100	37300	441	782	845	881	58	127	94	119	100	00050
1200	1190	83500	38800	446	780	849	861	53	129	90	139	103	00052
1250	1240	86900	40400	440	770	830	871	59	139	109	129	109	00055
1300	1290	90400	42000	421	745	780	839	78	164	159	161	141	00071

C Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C		
100	100	7500	3500	521	931	669	988	0	0	0	0	0	0
300	300	21400	10100	520	900	671	949	0	31	-02	39	17	.00009
500	500	35200	16600	500	879	660	920	21	51	09	68	37	00019
700	700	49100	23000	479	860	640	900	42	71	29	88	58	00029
900	900	63000	29700	467	838	629	854	54	93	4[illegible]	134	80	00040
950	950	66500	31200	464	850	620	839	57	81	49	149	84	00042
1000	1000	69900	32800	460	850	620	831	61	81	49	157	87	00044
1050	1050	73400	34500	451	852	615	821	70	79	54	167	93	00047
1100	1090	76600	36200	452	850	609	811	69	81	60	177	9[illegible]	00049
1150	1140	80100	37700	443	841	607	801	78	90	62	181	104	00052
1200	1190	83500	39400	449	833	603	788	72	98	66	200	09	00055
1250	1240	86900	40800	442	830	590	790	79	101	79	198	114	00057
1300	1290	90400	42600	427	815	531	790	94	116	138	198	137	00069

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
100	100	7500	3500	780	970	911	670	0	0	0	0	0	0
300	300	21400	10200	780	947	912	631	0	23	[illegible]	39	1[illegible]	00008
500	500	35200	16700	758	940	890	601	22	[illegible]	2[illegible]	69	36	00018
700	700	49100	23100	730	909	880	571	50	[illegible]	31	[illegible]	60	00030
900	900	63000	29800	712	895	862	548	68	75	49	122	79	00040
950	950	66500	31300	710	898	858	543	70	72	5[illegible]	127	8[illegible]	00041
1000	1000	69900	32900	689	878	840	539	91	92	71	131	96	00048
1050	1050	73400	34600	690	881	850	539	90	89	61	131	93	00047
1100	1090	76600	36300	690	890	845	524	90	80	65	146	9[illegible]	00048
1150	1140	80100	37800	681	887	839	521	99	83	72	149	01	00051
1200	1190	83500	39500	680	880	840	510	00	90	71	160	[illegible]	00053
1250	1240	86900	40900	676	875	830	511	102	95	8[illegible]	159	10[illegible]	00055
1300	1290	90400	42700	650	899	790	511	130	75	[illegible]	[illegible]	12[illegible]	00060

TABLE XXXVI

TUBE NO. 5

Axial Tension Only

Inside Diameter of Tube 5.554 inches

Circumferential Gage Lines

A Lines

D Lines

Machine Load lbs	Av Axial Unit Stress lbs sq.in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
		1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
5850	3600	70.0	52.5	60.0	52.0	0	0	-0	0	0	0
20600	12100	70.5	55.0	61.2	53.8	-0.5	-2.5	-1.2	-1.8	1.5	-.00008
30600	18900	71.2	56.0	64.7	56.1	-1.2	-3.5	-4.7	-4.1	-3.4	-.00017
42200	26000	73.1	57.8	66.5	58.3	-3.1	-5.3	-6.5	-6.3	-5.3	-.00021
50100	30900	74.0	57.6	66.0	58.0	-4.0	-5.1	-6.0	-6.0	-5.3	-.00027
61200	37700	76.9	59.0	69.0	61.5	-6.9	-6.5	-9.0	-9.5	[illegible]	-.00040
64600	39800	77.8	61.2	68.4	61.3	-7.8	-8.7	-8.4	-9.3	-8.6	-.00043
68100	42000	79.1	62.4	68.3	62.4	-9.1	-9.9	8.3	-10.4	-9.4	-.00047
72300	44600	78.9	63.1	71.2	63.0	-8.9	-10.6	-11.2	-11.0	.10.4	.00052
76100	46900	78.6	64.0	72.0	63.0	-8.6	-11.5	-12.0	-11.0	10.8	-.00054
79800	49200	77.9	66.2	73.0	65.5	-7.9	-13.7	-13.0	-13.5	-12.0	.00060
82400	50800	79.0	66.5	74.3	64.5	-9.0	-14.0	14.3	-12.5	-12.5	-.00063
85200	52500	78.1	65.9	75.0	67.5	-8.1	-13.4	-15.0	15.5	13.0	-.00065

TABLE XXXVI

TUBE NO. 5

Axial Tension Only

Inside Diameter of Tube 5.554 inches

Circumferential Gage Lines

A Lines

Machine Load lbs.	Av Axial Unit Stress lbs sq. in.	Reading on Gage Line 1-2-A	2-3-A	3-4-A	4-1-A	Differences 1-2-A	2-3-A	3-4-A	4-1-A	Average Difference	Average Unit Elongation
5850	3600	611	130	52.1	411	0	0	0	0	0	0
20600	12700	65.8	172	539	44.0	-47	-42	-1.8	-29	-3.4	-.00017
30600	18900	660	186	549	45.0	-49	5.6	-28	-39	-43	-00022
42200	26000	66.0	220	567	490	-49	9.0	-4.6	-7.9	-66	-.00033
50100	30900	683	200	57.5	494	-7.2	70	5.4	83	-70	-.00035
61200	37700	69.1	235	60.0	50.5	-80	-10.5	-79	[illegible]	-90	-00045
64600	39800	699	249	600	51.0	-88	-11.9	-79	-99	-96	[illegible]
68100	42000	70.8	258	59.0	51.9	-97	-128	-69	108	-101	[illegible]
72300	44600	728	277	61.1	530	117	-147	-9.0	-119	118	[illegible]
76100	46900	72.0	295	60.0	551	-10.9	-165	-79	-140	-123	-.00062
79800	49200	715	312	61.5	[illegible]	-10.4	-182	-94	140	[illegible]	[illegible]
82400	50800	740	325	60.0	539	-12.9	-19.5	-7.9	128	-133	00068
85200	52500	745	33.0	63.5	55.0	-13.4	-20.0	-114	139	-147	.00074

B Lines

Machine Load lbs.	Av. Axial Unit Stress lbs. sq. in.	Reading on Gage Line 1-2-B	2-3-B	3-4-B	4-1-B	Differences 1-2-B	2-3-B	3-4-B	4-1-B	Average Difference	Average Unit Elongation
5850	3600	68.9	82.1	43.1	40.0	0	0	0	0	0	0
20600	12700	702	84.9	44.5	439	-1.3	-28	-1.4	-3.9	-2.4	-00012
30600	18900	70.3	89.0	44.5	444	-1.4	-6.9	-1.4	-4.4	-3.5	-00018
42200	26000	73.1	92.0	46.2	47.0	-4.2	-9.9	-3.1	-7.0	-6.0	-.00030
50100	30900	729	93.3	460	47.2	-4.0	-11.2	-2.9	-7.2	-6.3	-.00032
61200	37700	736	94.2	46.9	49.1	-4.7	-12.1	-38	-9.1	-7.4	-00037
64600	39800	75.9	95.8	47.5	497	-7.0	-13.7	-44	-9.7	-87	-.00044
68100	42000	764	96.0	485	51.1	-7.5	-13.9	-5.4		-9.5	[illegible]
72300	44600	78.6	988	496	52.6	-97	-167	-65	-12.6	-11.2	-00057
76100	46900	78.5	3.8	490	53.2	-96	-217	-59	[illegible]	12.6	[illegible]
79800	49200	80.0	0.7	490	57.1	-11.1	-186	-59		-13.2	[illegible]
82400	50600	810	0.0	495	58.0	-12[illegible]	-179	-64		-13[illegible]	000[illegible]
85200	52500	843	1.0	522	59.0	-154	89	-9	-19.0		-00078

C Lines

Machine Load lbs	Av. Axial Unit Stress lbs sq. in.	Reading on Gage Line 1-2-C	2-3-C	3-4-C	4-1-C	Differences 1-2-C	2-3-C	3-4-C	4-1-C	Average Difference	Average Unit Elongation
5850	3600	51.8	320	703	671	0	0	0	0	0	[illegible]
20600	12700	525	340	705	695	-07	20	-02	-25	1.4	-000[illegible]
30600	18900	552	354	736	702	-3.4	-34	-33	[illegible]	[illegible]	00017
42200	26000	56.0	375	740	722	-42	5.5	-37	-52	-47	[illegible]024
50100	30900	580	301	748	733	-62	-6[illegible]	-45	-63	-58	000[illegible]
61200	37700	591	408	765	750	-73	-88	-[illegible]2	-80	70	[illegible]
64600	39800	60.1	409	763	770	-63	-89	-60	-100	[illegible]	[illegible]
68100	42000	61.6	41.1	762	756	-9.8	91	-59	-86	[illegible]4	00042
72300	44600	61.7	420	774	785	-99	100	-71	[illegible]	96	00048
76100	46900	63.5	445	795	796	-11.7	-125	-92	-126	115	-00058
79800	49200	630	444	790	821	-11.2	-124	-87	-15	-119	-00060
82400	50800	62.0	445	785	825	-102	125	-82	[illegible]	-11.6	.00058
85200	52500	620	446	800	825	-10.2	126	-97	-155	-12.0	00060

D Lines

Machine Load lbs	Av Axial Unit Stress lbs sq. in	Reading on Gage Line 1-2-D	2-3-D	3-4-D	4-1-D	Differences 1-2-D	2-3-D	3-4-D	4-1-D	Average Difference	Average Unit Elongation
5850	3600	70.0	525	600	520	0	0	-0	0	0	[illegible]
20600	12100	705	55.0	612	538	-05	-25	-12	-18	[illegible]5	-000[illegible]
30600	18900	712	560	647	561	-1.2	-35	-47	-4[illegible]	34	-00017
42200	26000	731	578	665	583	-3.1	-53	-65	-63	-53	-00021
50100	30900	74.0	576	660	580	-40	-51	-60	-60	[illegible]	[illegible]027
61200	37700	769	590	690	615	-69	-65	-90	-95	[illegible]	00040
64600	39800	778	612	684	613	-78	-87	84	-93	-86	[illegible]
68100	42000	791	624	683	624	-91	-99	83	-104	94	00047
72300	44600	789	631	712	630	-89	-106	-112	-110	-104	00052
76100	46900	786	64.0	720	630	-86	-115	-120	-11.0	108	-0005[illegible]
79800	49200	779	662	730	635	-7.9	-137	130	-135	120	000[illegible]0
82400	50800	790	665	743	645	-90	-140	-14[illegible]	-125	125	-000[illegible]
85200	52500	78[illegible]	659	750	675	-81	-134	-15.0	155	-130	[illegible]065

TABLE XXXVII.

TUBE NO 6

Ratio of Hoop to Axial Tension 0.92
Inside Diameter of Tube 5.579 inches
Circumferential Gage Lines.

A Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg.	Corrected	lbs.	lbs. sq. in.	1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A	Difference	Unit Elongation
100	95	5700	3200	3.8	9.0	11.0	93.1	0	0	0	0	0	0
300	285	16000	9700	1.5	6.3	7.5	92 0	2.3	2.7	3.5	1.1	2 4	.00012
500	475	27200	16100	98.5	4.2	4.8	91.0	5.3	4.8	6.8	2.1	4.8	.00024
600	575	31500	19400	97.1	3.9	3.1	90.1	6.7	5.1	7.9	3.0	5 7	.00029

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs.	lbs sq. in	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Unit Elongation
100	95	5700	3300	18.0	28 9	20.0	25.0	0	0	0	0	0	.0
300	285	16000	9900	18 8	24.5	21.0	21.0	-0.8	4.4	-1.0	4.[illegible]	1.7	.00009
500	475	27200	16600	16 9	20.8	20.9	18.9	1.1	8.1	-0.9	6.1	3.6	.00018
600	575	31500	20000	16.1	19.1	19 3	16.8	1 9	9.8	0.7	8 2	5.2	.00026
700	670	36600	23300	13.1	16.1	19 0	14.0	4 9	12 8	1.0	11.0	7.4	.00037
750	720	39200	25000	12 1	16.0	18 4	12.4	5 9	12.9	1.6	12.6	8.3	.00042
800	765	41700	26600	12.0	14 2	18 3	10 8	6.0	14.7	1.7	14.2	9.2	.00046
850	815	44300	28300	11 0	11 0	18.0	8.0	7 0	17.9	2.0	17.0	11.0	.00055
900	865	47000	30000	11 5	7 0	18 0	5.0	6.5	21.9	2.0	20.0	12.6	.00063
950	915	49600	31700	3.3	97 0	12.0	99.0	14.7	31.9	8.0	26.0	20.2	.00101

TABLE XXXVII.

TUBE NO 6

Ratio of Hoop to Axial Tension 0.92
Inside Diameter of Tube 5.579 inches
Circumferential Gage Lines

A Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg.	Corrected	lbs.	lbs sq in.	1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A	Difference	Unit Elongation
100	95	5700	3200	3.8	9.0	11.0	93.1	0	0	0	0	0	0
300	285	16000	9700	1.5	6.3	7.5	92.0	23	2.7	3.5	1.1	24	.00012
500	475	27200	16100	98.5	4.2	4.8	91.0	5.3	4.8	6.8	2.1	48	.00024
600	575	31500	19400	97.1	3.9	3.1	90.1	6.7	5.1	7.9	3.0	57	.00029
700	670	36600	22500	96.0	1.1	1.0	88.1	7.8	7.9	10.0	5.0	77	.00039
750	720	39200	24200	95.2	0.0	0.0	86.9	8.6	9.0	11.0	6.2	8.7	.00044
800	765	41700	25800	94.8	99.1	99.0	86.1	9.0	9.9	12.0	7.0	9.5	.00048
850	815	44300	27400	94.8	95.9	97.5	84.5	9.0	13.1	13.5	8.6	11.1	.00056
900	865	47000	29100	94.1	95.5	95.6	83.0	9.7	13.5	15.4	10.1	12.1	.00061
950	915	49600	30700	88.0	84.7	92.2	78.1	15.8	24.3	18.8	15.0	18.5	.00093

B Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg.	Corrected	lbs.	lbs. sq. in	1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B	Difference	Unit Elongation
100	95	5700	3200	13.1	3.1	18.9	10.5	0	0	0	0	0	0
300	285	16000	9700	12.1	1.3	15.9	8.1	1.0	1.8	3.0	2.4	2.1	.00011
500	475	27200	16100	8.2	98.4	15.0	7.0	4.9	4.7	3.9	3.5	4.3	.00022
600	575	31500	19400	8.3	98.1	14.1	5.2	4.8	5.0	4.8	5.3	5.0	.00025
700	670	36600	22500	7.0	95.1	13.3	4.1	6.1	8.0	5.6	6.4	6.5	.00033
750	720	39200	24200	6.5	93.9	13.0	3.0	6.6	9.2	5.9	7.5	7.3	.00037
800	765	41700	25800	7.0	92.2	12.9	1.1	6.1	10.9	6.0	8.4	7.9	.00040
850	815	44300	27400	5.2	90.0	11.0	99.8	7.9	13.1	7.9	10.7	9.9	.00050
900	865	47000	29100	4.5	89.0	11.1	96.7	8.6	14.1	7.8	13.8	11.1	.00056
950	915	49600	30700	99.1	86.3	6.1	92.1	14.0	16.8	12.8	18.4	15.5	.00078

C Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs.	lbs. sq. in.	1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C	Difference	Unit Elongation
100	95	5700	3200	36.7	0.9	27.0	10.0	0	0	0	0	0	0
300	285	16000	9800	35.0	98.0	25.8	6.9	1.7	2.9	1.2	3.1	2.2	.00011
500	475	27200	16400	34.0	94.3	25.0	4.2	2.7	6.6	2.0	5.8	4.3	.00022
600	575	31500	19700	33.0	93.2	24.2	2.2	3.7	7.7	2.8	7.8	5.5	.00028
700	670	36600	22900	32.1	90.8	23.2	1.0	4.6	10.8	3.8	9.0	7.1	.00036
750	720	39200	24600	32.0	88.2	22.3	99.0	4.7	12.7	4.7	11.0	8.3	.00042
800	765	41700	26200	31.9	86.0	21.4	97.0	4.8	14.9	5.6	13.0	9.6	.00048
850	815	44300	27800	30.4	84.0	20.3	95.0	6.3	16.9	6.7	15.0	11.2	.00056
900	865	47000	29600	30.0	82.0	19.5	90.1	6.7	18.9	7.5	19.9	13.0	.00065
950	915	49600	31200	21.0	75.0	18.0	84.0	15.7	25.9	9.0	26.0	19.2	.00096

D Lines

Water Pressure		Total Load	Average Unit Stress	Reading on Gage Line				Differences				Average	Average
Gage Rdg	Corrected	lbs	lbs. sq. in	1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D	Difference	Unit Elongation
100	95	5700	3300	18.0	28.9	20.0	25.0	0	0	0	0	0	0
300	285	16000	9900	18.8	24.5	21.0	21.0	-0.8	4.4	-1.0	4	1.7	.00009
500	475	27200	16600	16.9	20.8	20.9	18.9	1.1	8.1	-0.9	6.1	3.6	.00018
600	575	31500	20000	16.1	19.1	19.3	16.8	1.9	9.8	0.7	8.2	5.2	.00026
700	670	36600	23300	13.1	16.1	19.0	14.0	4.9	12.8	1.0	11.0	7.4	.00037
750	720	39200	25000	12.1	16.0	18.4	12.4	5.9	12.9	1.6	12.6	8.3	.00042
800	765	41700	26600	12.0	14.2	18.3	10.8	6.0	14.7	1.7	14.2	9.2	.00046
850	815	44300	28300	11.0	11.0	18.0	8.0	7.0	17.9	2.0	17.0	11.0	.00055
900	865	47000	30000	11.5	7.0	18.0	5.0	6.5	21.9	2.0	20.0	12.6	.00063
950	915	49600	31700	3.3	97.0	12.0	99.0	14.7	31.9	8.0	26.0	20.2	.00101

TABLE XXXVIII.

TUBE NO. 7

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.588 inches

Circumferential Gage Lines

TABLE XXXVIII.

TUBE NO. 7

Ratio of Hoop to Axial Tension 0.475

Inside Diameter of Tube 5.588 inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
100	95	10300	3100	37.1	93.0	73.1	65.2	0	0	0	0	0	0
200	190	20000	6000	340	979	721	65.8	3.1	0.1	1.0	- 0.6	0.9	.00005
300	285	29700	8500	32.3	96.9	71.3	66.0	4.8	1.1	1.8	- 0.8	1.7	.00009
400	380	39300	11900	32.5	94.0	70.7	65.8	4.6	4.0	2.4	- 0.6	2.7	.00014
450	430	44100	13300	32.2	93.2	71.5	66.6	4.9	4.8	1.6	- 1.4	2.5	.00013
500	475	48900	14800	31.9	939	699	65.6	5.2	4.1	3.2	-0.4	3.0	.00015
550	525	53900	16200	32.3	92.9	71.2	64.5	4.8	5.1	1.9	+0.7	3.1	.00016

B Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-B	2-3-B	3-4-B	4-1-B	1-2-B	2-3-B	3-4-B	4-1-B		
100	95	10300	3100	69.1	400	51.3	84.0	0	-0	0	0	0	0
200	190	20000	6000	68.0	402	51.0	84.6	1.1	-0.2	0.3	-0.6	0.2	.00001
300	285	29700	8500	66.1	398	50.1	84.9	3.0	+0.2	1.2	-0.9	0.9	.00005
400	380	39300	11900	64.9	38.0	49.6	84.1	4.2	2.0	1.7	-0.1	2.0	.00010
450	430	44100	13300	64.4	38.9	49.5	84.9	4.7	1.1	1.8	-0.9	1.7	.00009
500	475	48900	14800	60.7	37.5	46.9	85.0	8.4	2.5	4.4	-1.0	3.6	.00018
550	525	53900	16200	60.1	36.0	49.7	85.0	9.0	4.0	1.6	-1.0	3.4	.00017

C Lines

Water Pressure		Total Load lbs	Average Unit Stress lbs sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C		
100	95	10300	3100	72.0	43.7	99.9	99.0	0	0	0	0	0	0
200	190	20000	6000	70.0	43.7	99.0	98.3	2.0	0	0.9	0.7	0.9	.00005
300	285	29700	8500	68.1	43.4	97.9	99.0	3.9	0.3	2.0	0	1.6	.00008
400	380	39300	11900	67.4	44.3	98.0	96.9	4.6	-0.6	1.9	2.1	2.0	.00010
450	430	44100	13300	68.0	44.5	97.0	95.0	4.0	-0.8	2.9	4.0	2.5	.00013
500	475	48900	14800	70.8	43.9	97.9	93.0	1.2	-0.2	2.0	6.0	2.3	.00012
550	525	53900	16200	70.0	43.0	99.0	94.8	2.0	+0.7	0.9	4.2	1.5	.00008

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
100	95	10300	3100	58.0	62.0	74.6	47.2	0	0	0	0	0	0
200	190	20000	6000	56.5	61.2	74.1	46.4	1.5	0.8	0.5	0.8	0.9	.00005
300	285	29700	8500	55.1	61.1	72.9	46.0	2.9	0.9	1.7	1.2	1.7	.00008
400	380	39300	11900	54.2	69.1	73.0	43.0	3.8	1.9	1.6	4.2	2.9	.00015
450	430	44100	13300	54.8	60.0	76.0	41.9	3.2	2.0	-1.4	5.3	2.3	.00012
500	475	48900	14800	55.8	57.9	77.0	39.4	2.2	4.1	-2.4	7.8	2.9	.00015
550	525	53900	16200	54.5	59.0	76.3	40.9	3.5	3.0	-1.7	6.3	2.8	.00014

TABLE XXXIX.

TUBE NO. 8

Ratio of Hoop to Axial tension 0.49

Inside Diameter of Tube 5.560 inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2 A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
100	100	5850	3200	2.0	95.5	1.7	8.0	0	0	0	0	0	0
400	400	21600	11800	97.8	92.4	98.9	6.5	4.2	3.1	2.8	1.5	2.9	.00015

TABLE XXXIX.

TUBE NO. 8

Ratio of Hoop to Axial tension 0.49

Inside Diameter of Tube 5.560 inches

Circumferential Gage Lines

A Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2 A	2-3-A	3-4-A	4-1-A	1-2-A	2-3-A	3-4-A	4-1-A		
100	100	5850	3200	2.0	95.5	1.7	8.0	0	0	0	0	0	0
400	400	21600	11800	97.8	92.4	98.9	6.5	4.2	3.1	2.8	1.5	2.9	00015
700	700	37300	20300	94.0	88.4	95.0	2.5	8.0	7.1	6.7	5.5	6.8	.00034
1000	1000	53100	28900	91.0	84.4	86.4	94.7	11.0	11.1	15.3	13.3	12.7	.00064
1100	1100	57900	31700	89.8	82.1	83.3	87.3	12.2	13.4	18.4	20.7	16.2	00081
1200	1200	63100	34400	81.4	71.0	76.0	82.0	20.6	24.5	25.7	26.0	24.2	0012[illegible]

B Lines

Water Pressure		Total Load lbs	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-B	2-3-B	3-4-B	4-1-B	1-2 B	2-3-B	3-4-B	4-1 B		
100	100	5850	3200	93.9	84.0	51.0	14.5	0	0	0	0	0	0
400	400	21600	11800	91.9	80.5	48.8	13.5	2.0	3.5	2.2	1.0	2.2	[illegible]
700	700	37300	20300	89.8	77.1	45.3	8.1	4.1	6.9	5.7	6.4	[illegible]	00029
1000	1000	53100	28900	83.0	74.0	39.0	3.0	10.9	10.0	12.0	11.5	11.1	00056
1100	1100	57900	31700	81.0	71.5	36.5	98.0	12.9	12.5	14.5	16.5	[illegible]	[illegible]
1200	1200	63100	34400	75.0	62.5	28.0	85.9	18.9	21.5	23.0	28.6	23.0	[illegible]

C Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-C	2-3-C	3-4-C	4-1-C	1-2-C	2-3-C	3-4-C	4-1-C		
100	100	5850	3200	91.0	19.0	97.5	28.1	0	0	0	0	0	0
400	400	21600	11800	87.5	15.4	95.0	23.9	3.5	3.6	2.5	4.2	3.5	00018
700	700	37300	20300	87.2	12.9	94.6	18.0	3.8	6.1	2.9	10.1	[illegible]	[illegible]
1000	1000	53100	28900	84.0	9.5	90.0	11.3	7.0	9.5	7.5	16.8	10.2	.00051
1100	1100	57900	31700	81.0	3.5	87.1	7.0	10.0	15.5	10.4	21.1	14.3	00[illegible]
1200	1200	63100	34400	75.2	98.0	78.3	0.1	15.8	21.0	15.[illegible]	[illegible]	[illegible]00	.00100

D Lines

Water Pressure		Total Load lbs.	Average Unit Stress lbs. sq. in.	Reading on Gage Line				Differences				Average Difference	Average Unit Elongation
Gage Rdg.	Corrected			1-2-D	2-3-D	3-4-D	4-1-D	1-2-D	2-3-D	3-4-D	4-1-D		
100	100	5850	3200	80.1	6.8	5.0	14.0	0	0	0	0	0	0
400	400	21600	11800	78.0	3.2	2.3	10.9	2.1	3.6	2.7	3.1	2.9	00015
700	700	37300	20300	75.2	0.0	0.3	6.8	4.9	6.8	4.7	7.7	5.9	00030
1000	1000	53100	28900	72.3	96.3	97.2	98.0	7.8	10.5	7.8	16.0	10.5	00053
1100	1100	57900	31700	70.2	92.8	95.5	94.3	9.9	14.0	9.5	19.7	13.3	00067
1200	1200	63100	34400	57.3	82.0	73.0	88.0	12.8	24.8	[illegible]	26.0	23.9	.00120

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VITA

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